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Alex. C. Humphreys

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OF
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS

TRANSACTIONS

VOLUME 34

CLEVELAND MEETING
NEW YORK MEETING
1912

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1913

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J. F. Holloway	1885	Died September 1, 1896
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George H. Babcock	1887	Died December 16, 1893
Horace See	1888	Died December 14, 1909
Henry R. Towne	1889	New York, N. Y.
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Robert W. Hunt	1891	Chicago, Ill.
Charles H. Loring	1892	Died February 5, 1907
Eckley B. Cox	1892-1894	Died May 13, 1895
E. F. C. Davis	1895	Died August 6, 1895
Charles E. Billings	1895	Hartford, Conn.
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Worcester R. Warner	1897	Cleveland, O.
Charles Wallace Hunt	1898	Died March 27, 1911
George W. Melville	1899	Died March 17, 1912
Charles H. Morgan	1900	Died January 10, 1911
S. T. Wellman	1901	Cleveland, O.
Edwin Reynolds	1902	Died February 19, 1909
James M. Dodge	1903	Philadelphia, Pa.
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INSTITUTION	DATE AUTHORIZED BY COUNCIL	HONORARY CHAIRMAN	CHAIRMAN	CORRESPONDING SECRETARY
Armour Inst. of Tech. Chicago, Ill.	Mar. 9, 1909	G. F. Gebhardt	E. R. Burley, Jr.	H. R. Kuehn
Brooklyn Poly Inst. Brooklyn, N. Y.	Mar. 9, 1909	W. D. Ennis	B. L. Huestis	A. Bielek
Columbia University New York	Nov. 9, 1909	Chas. E. Lucke	E. M. Stone	E. A. Jareckie
Cornell University Ithaca, N. Y.	Dec. 4, 1908	R. C. Carpenter	S. D. Mills	D. S. Wegg, Jr.
Lehigh University South Bethlehem, Pa.	June 2, 1911	H. A. S. Howarth	E. E. Finn	J. F. Beers, Jr.
Leland Stanford Jr. Univ., Palo Alto, Cal.	Mar. 9, 1909	W. F. Durand	C. W. Scholefield	V. W. Winters
Mass. Inst. of Tech. Boston, Mass.	Nov. 9, 1909	E. F. Miller	J. G. Russell	J. B. Farwell
New York University New York	Nov. 9, 1909	C. E. Houghton		
Ohio State University Columbus, Ohio	Jan. 10, 1911	Wm. T. Magruder	R. H. Neilan	R. M. Powell
Penna. State College State College, Pa.	Nov. 9, 1909	J. P. Jackson	J. F. Blank	G. W. Barger
Purdue University Lafayette, Ind.	Mar. 9, 1909	G. A. Young	A. W. Kimmel	G. F. Lynde
Rensselaer Poly. Inst. Troy, N. Y.	Dec. 9, 1910	A. M. Greene, Jr.	E. Kneass	R. F. Fox
State Univ. of Ky. Lexington, Ky.	Jan. 10, 1911	F. P. Anderson	R. R. Taliaferro	F. J. Forsyth
Stevens Inst. of Tech. Hoboken, N. J.	Dec. 4, 1908	Alex. C. Humphreys	J. H. Vander Veer	J. Strauss
Syracuse University Syracuse, N. Y.	Dec. 3, 1912	W. E. Ninde	O. W. Sanderson	R. A. Sherwood
Univ. of Arkansas Fayetteville, Ark.	Apr. 12, 1910	B. N. Wilson	M. McGill	C. Bethel
Univ. of California Berkeley, Cal.	Feb. 13, 1912	Joseph N. Le Conte	J. F. Ball	G. H. Hagar
Univ. of Cincinnati Cincinnati, Ohio	Nov. 9, 1909	J. T. Faig	C. F. Lytle	A. O. Hurxthal
Univ. of Illinois Urbana, Ill.	Nov. 9, 1909	W. F. M. Goss	L. G. Smith	C. A. Schoessel
University of Kansas Lawrence, Kan.	Mar. 9, 1909	F. W. Sibley	E. A. VanHouten	L. E. Knerr
Univ. of Maine Orono, Me.	Feb. 8, 1910	Arthur C. Jewett	E. H. Bigelow	O. H. Davis
Univ. of Missouri Columbia, Mo.	Dec. 7, 1909	H. Wade Hibbard	F. I. Kemp	R. Runge
Univ. of Nebraska Lincoln, Neb.	Dec. 7, 1909	J. D. Hoffman	P. S. Toney	M. C. Evans
Univ. of Wisconsin Madison, Wis.	Nov. 9, 1909	A. G. Christie	W. D. Moyer	W. K. Fitch
Washington University St. Louis, Mo.	Mar. 10, 1911	E. L. Ohle	D. Southerland	A. Schleiffarth
Yale University New Haven, Conn.	Oct. 11, 1910	L. P. Breckenridge	C. E. Booth	O. D. Covell

SUMMARY OF MEMBERSHIP

United States

Alabama	26	Nevada	3
Alaska	2	New Hampshire	14
Arizona	7	New Jersey	214
Arkansas	2	New Mexico	4
California	113	New York	1209
Colorado	32	North Carolina	17
Connecticut	169	North Dakota	3
Delaware	21	Ohio	296
District of Columbia	36	Oklahoma	2
Florida	2	Oregon	12
Georgia	22	Pennsylvania	520
Hawaii	6	Philippine Islands	2
Idaho	5	Porto Rico	5
Illinois	300	Rhode Island	74
Indiana	68	South Carolina	6
Iowa	15	South Dakota	1
Kansas	12	Tennessee	15
Kentucky	5	Texas	25
Louisiana	29	Utah	9
Maine	19	Vermont	17
Maryland	39	Virginia	35
Massachusetts	430	Washington	22
Michigan	151	West Virginia	9
Minnesota	30	Wisconsin	116
Mississippi	3	Wyoming	2
Missouri	78		
Montana	7		
Nebraska	7	Total	4268

Foreign Countries

Africa	14	Holland	1
Australia	10	India	4
Austria	2	Italy	3
Belgium	6	Japan	5
Canada	68	Mexico	12
Central America	12	Russia	4
Channel Islands	1	Scotland	2
China	3	South America	16
Cuba	6	Spain	3
England	57	Sweden	5
Federated Malay States	1	Switzerland	3
Finland	2	Turkey	1
France	11		
Germany	17	Total	269

SUMMARY OF MEMBERSHIP

By Residence

December 31, 1912

Membership in United States	4268
Membership in foreign countries	269
Present address unknown	5
Total membership	4542

By Grades

Honorary Members	19
Members	3098
Associates	347
Juniors	1078
Total membership	4542

GAS POWER SECTION

SUMMARY OF MEMBERSHIP

United States

Alabama	3	New Hampshire	1
Arkansas	1	New Jersey	13
California	9	New York	141
Connecticut	8	Ohio	22
Delaware	1	Oklahoma	1
District of Columbia	3	Oregon	2
Georgia	3	Pennsylvania	40
Illinois	20	Rhode Island	6
Indiana	11	Texas	1
Iowa	2	Utah	1
Maine	3	Vermont	1
Maryland	2	Virginia	1
Massachusetts	22	Washington	2
Michigan	13	West Virginia	2
Minnesota	5	Wisconsin	17
Missouri	7		
Nebraska	1	Total	364

Foreign Countries

Belgium	1	Japan	1
Canada	3	Mexico	1
England	1	South America	2
France	1		
Germany	2	Total	12

By Residence

Membership in United States	364
Membership in foreign countries	12
Address unknown	1
Total membership	377

By Grades

Members of the Society	263
Affiliates	114
Total	377

STUDENT BRANCHES

Armour Institute of Technology	13
Columbia University	37
Cornell University	205
Lehigh University	14
Leland Stanford Jr. University	16
Massachusetts Institute of Technology	51
Ohio State University	27
Pennsylvania State College	23
Polytechnic Institute of Brooklyn	17
Purdue University	25
Rensselaer Polytechnic Institute	32
State University of Kentucky	57
Stevens Institute of Technology	76
University of Arkansas	2
University of California	14
University of Cincinnati	46
University of Illinois	62
University of Kansas	19
University of Maine	17
University of Missouri	39
University of Nebraska	10
University of Wisconsin	9
Washington University	16
Yale University	40
Total	867

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TRANSACTIONS

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOLUME 34—1912

This volume comprises a record of the proceedings of The American Society of Mechanical Engineers for 1912 and is mainly devoted to the two general meetings of the Society known as the Spring and Annual Meetings. Brief reference is made to the various local meetings held throughout the year and in cases where papers presented locally have later been assigned to the Spring or Annual Meetings they appear in this volume. An account of the main events connected with the work of the Society is given in the annual report of the Council, beginning on page 4. The reports of the standing committees and a more complete account of many of the activities will be found in THE JOURNAL, issued monthly by the Society.

ALEXANDER CROMBIE HUMPHREYS

At the Annual Meeting 1911, Alexander Crombie Humphreys was elected President of The American Society of Mechanical Engineers for the ensuing year, and the work of his administration is comprised in this volume.

Dr. Humphreys, who is the second president of Stevens Institute of Technology, was born in Edinburgh, March 30, 1851. He came to this country at the age of eight and was educated in his father's private school, Boston, Mass. When fourteen years old he passed a preliminary test examination for the United States Naval Academy, but was disqualified on account of his youth. He then entered a Boston insurance office. In 1866 he re-

moved to New York, where he secured employment with the New York Guaranty & Indemnity Company, and in a few years was made receiving teller and assistant bookeeper. In 1872 he became secretary-treasurer, and shortly afterwards superintendent of the Bayonne & Greenville Gas Light Company. For years he had spent much of his spare time working with tools, and he now came in touch with mechanical undertakings as secretary of the building committee of his company. Upon the completion of the plant he was offered the position of secretary and treasurer. Three years later he recognized his need of a technical course at Stevens Institute, and presented himself at recitations and lectures two mornings of each week, doing the prescribed work of the institute at home. During the next four years he kept up with his class, attended to his business duties and supported his wife and two children. In 1881 the president and the faculty of the institute passed a formal resolution congratulating him upon his success under these exceptional conditions. Shortly after his graduation he accepted the office of chief engineer of the Pintsch Lighting Company, for which he built many oil-gas works, conducted experimental work on a large scale, and perfected an organization for carrying on the business. In January 1885, he became superintendent of construction of the United Gas Improvement Company of Philadelphia, and within a few months was its general superintendent and chief engineer. Under a uniform system of management and control developed by him, many properties were operated from Philadelphia. Early in 1892, in association with Arthur G. Glasgow, he established the London firm of Humphreys & Glasgow, consulting gas engineers, which was successful from the start. This firm designed and constructed the carbureted water gas plant. Other works constructed by Humphreys & Glasgow are now in operation in all parts of the world except North America. Here Mr. Humphreys built many plants as chief engineer of the United Gas Improvement Company. Upon his resignation from the United Gas Improvement Company, two years later, he formed the New York firm of Humphreys & Glasgow, which has been equally prominent in the American field. The specialty of gas engineering presents technical difficulties that few people appreciate, but since entering the field of consultant practice Mr. Humphreys has achieved a reputation far beyond the confines of this branch. When the presi-

dency of Stevens Institute was rendered vacant by the death of Henry Morton, the faculty unanimously petitioned the Board of Trustees to elect Mr. Humphreys, and this action was seconded by many of the alumni individually, who urged his appointment as a man of broad sympathies, an able administrator, and a thorough man of affairs. He was elected June 5, 1902. As president of this great school of practical knowledge he is doing more good than can be readily measured; his outspoken criticisms of technical education, as viewed in some quarters, have already caused a profound conviction of the need of more thoroughness and less display. The University of Pennsylvania has since conferred upon Mr. Humphreys the honorary degree of Sc.D., and the honorary degree of LL.D. has been conferred upon him by Columbia University, New York University and Princeton University.

In 1907 Dr. Humphreys succeeded Bayard Dod as president of the Board of Trustees of Stevens Institute. He is now president of the Buffalo Gas Company. He is a member of the American Institute of Mining Engineers, of the American Institute of Electrical Engineers, the American Society of Civil Engineers, the Institution of Civil Engineers, the British Association for the Advancement of Science, and the American Association for the Advancement of Science; past-president of the American Gas Light Association and of the American Gas Institute, the Robert Burns Society, New York, the Canadian Society, New York, and the National Society for the Promotion of Industrial Education; president of the Engineers' Club of New York; a member of the National Society for the Promotion of Engineering Education, and of the Union League, Century, Lotos, Lawyers' and Chemists' Clubs of New York; vice-president of the St. Andrew's Society, New York; a member of the University Club, Philadelphia, and many other societies working in public interest; a member of the Chamber of Commerce, New York, and president of Humphreys & Miller, Inc., New York, successors to Humphreys & Glasgow.

ANNUAL REPORT OF THE COUNCIL

The activities of the Society for the year 1911-1912 have covered a wide scope. Prominent among them has been coöperation with other societies. For this purpose a conference committee has been formed, consisting of two appointees each from the American Society of Civil Engineers, the American Institute of Mining Engineers, the American Institute of Electrical Engineers and The American Society of Mechanical Engineers, whose province it is to bring to the attention of their respective organizations for action matters of importance. The Society has also coöperated with other organizations and appointed committees to assist.

A Committee on Engineering Standards has been formed, with which the prominent engineering societies of the world have been asked to coöperate, to establish a clearing house for the issuance of information respecting standards for the engineering profession.

At various functions during the year, the Society has been represented by the following Honorary Vice-Presidents, appointed by the Council: William M. Rice Institute, Houston, Texas, W. B. Tuttle; Fourth National Conservation Congress, George M. Brill and W. B. Jackson; American Electric Railway Association, R. W. Hunt and C. J. Davidson; International Rubber Conference, E. B. Katte; International Conference on Hygiene and Demography, H. de B. Parsons and T. Kennard Thompson; Pratt Institute Commencement Exercises, C. W. Obert; Carnegie Institute of Technology, O. P. Hood; National Waterways Commission, H. G. Stott; International Congress of Navigation, Wm. T. Donnelly; American Academy of Political and Social Science, Geo. W. Melville, C. M. Day, J. M. Rusby.

Ties with organizations abroad as well as at home are continually being strengthened by the Society's policy of coöperation. During the year the Society has participated in the erec-

tion of a memorial window at Westminster Abbey to the memory of Lord Kelvin, under the auspices of the Institution of Civil Engineers, $600 having been subscribed by members of the Society for this purpose. In April of this year, the members of the German Museum Commission appointed to study important engineering achievements in America, were given the greetings of the Society at a meeting in the Engineering Societies Building, and in the various cities visited were assisted by committees of the membership formed for the purpose. An invitation to participate in a meeting of the Verein deutscher Ingenieure in Leipzig in the summer of 1913 has been accepted by the Council and active preparations for the trip have been begun by the Committee of Arrangements, Col. E. D. Meier, chairman.

Within the Society itself new activities have been undertaken. As a result of an earnest consideration by the Council of ways and means of extending the membership, a committee on Increase of Membership, I. E. Moultrop, chairman, was appointed early in the year. This committee has coöperated with the Membership Committee in seeking to invite only those qualified for admission from among those prominent in the profession and if possible increasing rather than lowering in any way the present high standard of membership. The proposed amendments to the Constitution provide an additional engineering grade, making the several grades of membership similar to the more important American and English societies and simplifying the method of election in accordance with these same prominent organizations. The proposed engineering grade is that of Associate-Member, intermediate between Member and Junior, and election by the Council is proposed to be substituted for election by the membership.

Honorary membership has been conferred during the year upon Anatole Mallet of Paris, Carl Gustav Patrick DeLaval, Stockholm, Rudolph Diesel, Munich, and Oskar von Miller, Munich. The death of George Wallace Melville, Past-President and Honorary Member, is reported with regret.

The following table shows the changes in membership for the fiscal year, October 1, 1911, to October 1, 1912, in distinction from the administration year which the rest of the report covers:

MEMBERSHIP FOR FISCAL YEAR (OCT. 1, 1911–OCT. 1, 1912)

Grade	Oct. 1, 1911	Losses				Additions			Net decrease	Net increase	Oct. 1, 1912
		Transfer	Resignations	Lapsed	Death	Transfer	Election	Reinstatements			
Honorary.....	16	..	..	..	1	..	3	..	..	2	18
Members.....	2702	1	19	30	27	54	278	4	..	259	2961
Associates.....	370	13	4	10	4	2	16	..	13	..	357
Juniors.....	919	43	17	28	2	1	175	1	..	87	1006
Total.....	4007	57	40	68	34	57	472	5	..	335	4342
Affiliates Gas Power Section.	134	.....				.....			8	..	126
Affiliates Student Section..	635	.....				.....			..	87	722

The University of California has been added to the list of student branches, which now number 25. The total number of student affiliates is now 722. The Committee on Student Branches, Prof. F. R. Hutton, chairman, has been increased by the addition of Wm. Kent and George A. Orrok, with the intent that through this enlarged committee visits can be made to the several branches.

An important step with reference to local meetings of the Society was taken at the Spring Meeting in Cleveland, after conference with representatives of the several cities where meetings are held. The policy adopted was to extend the rules for professional sections making them applicable to geographical sections. Upon approval of the Council local meetings may now be considered geographical groups, with the local committee in responsible charge. A close relation between the Society's headquarters and those of the several centers is maintained by the interchange of papers. Under these provisions meetings as geographical groups are now being conducted in San Francisco, St. Louis and Cincinnati.

The Council reports the final adoption of amendments to the By-Laws and Rules, as follows:

B 18 The Council at any meeting may, in its discretion, permanently remit the dues of any Member of the Society who has paid thirty years' dues, or who shall have reached the age of seventy years after having paid

twenty-five years' dues, provided that notice of such proposed action shall have been given at a previous meeting of the Council and the Committee on Membership shall have concurred in recommending that this action be taken. The Council may, in its discretion, restore to membership any person dropped from the rolls for non-payment of dues, or otherwise, upon such terms and conditions as it may at the time deem best for the interests of the Society.

R 13 The features of the program outside of the professional and business sessions to be provided for the members and guests at any meeting of this Society in any city shall be in charge of a Local Committee, subject, however, to the general approval of the Committee on Meetings. The President's reception of the annual meeting shall be in charge of the House Committee under the general supervision of the Committee on Meetings.

PROFESSIONAL SECTIONS AND GEOGRAPHICAL GROUPS

R 19 A professional section or a geographical group of the Society shall consist of Honorary Members, Members, Associates, and Juniors of The American Society of Mechanical Engineers and of other persons to be designated Affiliates as hereinafter described.

R 20 A professional section of The American Society of Mechanical Engineers may, with the approval of the Council, be organized for the consideration of any engineering, scientific, or professional topic, provided that a number satisfactory to the Council, of members of The American Society of Mechanical Engineers, unite in making written request for such an organization. A similar procedure shall apply to geographical groups. Such a section or group shall be designated as —— Section or —— Group of The American Society of Mechanical Engineers,—the blank being filled by the topic specialized or name of locality.

R 21 The provisions of the Constitution, By-Laws and Rules of The American Society of Mechanical Engineers, and the precedents of the Society with respect to professional sessions for the discussion of papers, shall cover the procedure of the professional sections or geographical groups except that no meeting of a section or group shall be considered a meeting of the Society as a whole.

R 22 For the convenient conduct of its professional affairs the section or group shall organize an Executive Committee of five members of the Society, under the general direction of the Council. Such officers as the section or group shall require must be selected from the membership of the Society. Other committees of the section or group shall be appointed by its Executive Committee.

R 23 The Executive Committee of the section or group, subject to the approval and direction of the Secretary of the Society, shall designate a Secretary of the section or group, whose duties shall be those usually attaching to the Secretary of a professional session, and who shall also see that the discussions of papers are satisfactorily reported and transmitted to the Secretary of the Society.

R 24 Expenditures for the purpose of a section or group chargeable to the Society must be authorized by the Secretary of the Society before they

are incurred, and must be provided for in the estimate and budget approved by the Council. No liability otherwise incurred shall be binding on the Society. Any expenditure not so provided must be met by the section or group itself.

R 25 Engineers and others not members of The American Society, but desiring to participate in the meeting of the section or group, may enroll themselves as affiliates as heretofore provided, with the approval of the Executive Committee of the section or group. Such affiliates shall have the privilege of presenting papers and taking part in the discussions. They shall pay $5 per annum, which shall be due and payable in advance, on October 1st of each year of their enrollment, and shall thereby be entitled to receive the regular issues of The Journal for a period covered by their dues.

R 26 The Council of The American Society of Mechanical Engineers may, at sixty days notice, suspend or disband any section or group.

Committee activity has been marked throughout the year and the advance which the Society is making has come about through the earnest work of the standing and special committees. Full details appear in the separate reports of the committees published in The Journal for December, 1912.

The task of completing the series of pictures of Honorary Members has progressed under the care of the House Committee, and twenty-two of these portraits are now hung upon the walls of the Society's rooms. To this committee has been assigned the duties in connection with the President's Reception given each year at the Annual Meeting.

Through the initiative of the Library Committee amendments to the By-Laws of the United Engineering Society have been prepared and have received the approval of the Boards of each of the Societies comprising the United Engineering Society, whereby a common administration of the joint libraries of the American Institute of Mining Engineers, the American Institute of Electrical Engineers and The American Society of Mechanical Engineers, may be secured, making possible greater efficiency in the joint library service, together with symmetry of the several collections, so that there may be no duplication of effort.

It may be of interest to add that this policy has been in the mind of the Trustees from the start of the plans for the building in 1904. It was especially dear to the broad intelligence and ambitions of the late Charles Wallace Hunt, Past-President of the Society and one time President of the Board of Trustees, but he

saw that during his life time the time was not ripe for its accomplishment. It is pathetic that within a year after his death this should have come to fruition, and he not able to enjoy the satisfaction it would have afforded him.

An important advance movement has been undertaken by the Committee on Meetings in the appointment of six additional sub-committees, covering various phases of mechanical engineering, with the idea of securing increased service through the assistance of specialists who will report developments in their respective fields. There are now ten of these sub-committees, covering Administration, Air Machinery, Cement Manufacture, Fire Protection, Hoisting and Conveying, Industrial Building, Iron and Steel, Machine Shop Practice, Railroads, and Textiles.

Due to the rapid growth of the Society, the work of the Publication Committee has become increasingly difficult, particularly as regards Transactions, there being sufficient important material to fill a two volume edition. The Council has concurred in the recommendation of the committee that an index to Transactions, comprising volumes 1 to 30 inclusive, be published, and an appropriation has been secured for the purpose. The subscription price of The Journal has been reduced to $2 a year to members and affiliates and $3 a year to non-members. This publication has been entirely self-supporting during the year and has continually sought to enlarge its field of activity and usefulness, so as to present a complete record of mechanical engineering.

A total of 865 applications for membership have been received by the Membership Committee during the fiscal year, of which 694 have been recommended for membership. Eight former members have also been reinstated. The Committee have adopted a simpler form of application and reply blank which will more readily meet its requirements.

A special committee on Code of Ethics has been appointed by the Council, following the example of other societies, to consider the formation of such a code.

The special committee on Flanges has coöperated with the National Association of Master Steam and Hot Water Fitters in formulating the 1912 schedule of standard weight and extra-heavy flanges and flanged fittings. This has been adopted by the Government in several of its departments and by numerous organizations.

As a result of interest awakened in the subject of measurement of taps, bolts and screws, a committee has been appointed to take up the matter, whose duty it shall be to prescribe the permissible tolerances in the commercial manufacture of these products, including their measurement.

The committee on Standard Cross-Sections and Symbols will report at the Annual Meeting on standards for drawings to indicate the materials used.

Upon invitation of the Patent Law Association, a conference was held in Washington at which the Society was represented, together with several other national societies. The Committee on Patent Law, W. H. Blauvelt and B. F. Wood, has been continued so that it may attend any conferences which may arise and participate in any developments, reporting to the Society as the necessity may arise.

At the Annual Meeting the very complete and remarkable report of the Committee on Power Tests will be presented and thus bring up to date and extend the present reports of the Society on these subjects.

No. 1346

MEETINGS JANUARY-JUNE

MEETINGS PREVIOUS TO THE SPRING MEETING

NEW YORK, JANUARY 9

A " get-together " meeting for a statement of plans and policies by members of the various standing committees and of the Committee on Meetings in New York, and for discussion of these policies. A full account of the meeting was published in The Journal for February 1912.

BOSTON, JANUARY 15

Annual Engineers Dinner, under the auspices of the Boston Society of Civil Engineers, the Boston Section of the American Institute of Electrical Engineers, and of this society. Chas T. Main, Mem. Am. Soc. M. E., president of the Boston Society of Civil Engineers presided, and addresses were given by Ira N. Hollis, Mem. Am. Soc. M. E., president of the Engineers Club of Boston; General Hugh Bancroft, chairman of the Directors of the Port of Boston; Prof. W. C. Sabin, Dean of the Graduate School of Applied Science of Harvard; Dr. Richard C. Maclaurin, president of Massachusetts Institute of Technology; and President F. W. Hamilton of Tufts College. The greetings of the national engineering societies were extended by Admiral M. T. Endicott, president of the American Society of Civil Engineers, Col. E. D. Meier, past-president and honorary vice-president of The American Society of Mechanical Engineers, and Dr. Gano S. Dunn, president of the American Institute of Electrical Engineers. The Journal for February 1912 contains a more detailed account of this meeting.

ST. LOUIS, FEBRUARY 7

A Comparison of Air Breaking Systems for Urban Electric Railway Cars: Paper by Charles W. Young and Charles A. Hobein, of the United Railways Company, St. Louis.

An abstract of this paper and the discussion appear in The Journal for July 1912.

NEW YORK, FEBRUARY 13

Design and Mechanical Features of the California Gold Dredge: Paper by Robert E. Cranston. Published in this volume.

PHILADELPHIA, FEBRUARY 14

Address by Charles Y. Knight, on his invention, the Knight silent motor.

BOSTON, FEBRUARY 16

Aviation: Paper by Albert A. Merrill, embodying the results of his extended study and experiments in this field.

An abstract of this paper appears in The Journal for May 1912.

PHILADELPHIA, MARCH 7

The Electrification of Main Line Railways: Paper by William J. Clark of New York. The accomplishments of electric traction were described, with special reference to its importance in the moving of freight.

NEW YORK, MARCH 12

Taps and Screws: Paper by F. O. Wells. An abstract of both the paper and discussion appears in the July issue of The Journal.

BOSTON, MARCH 16

Results of Tests on the Discharge Capacity of Safety Valves: Paper by E. F. Miller, and addition by A. B. Carhart. Published in this volume.

PROVIDENCE, MARCH 26

Dinner of the Providence Association of Mechanical Engineers, affiliated with the Society. Prof. Ira N. Hollis extended the greetings of the Society and Calvin W. Rice spoke briefly of the Society's activities. E. L. Corthell, the speaker of the evening, gave an address on the Future of South America.

PHILADELPHIA, MARCH 30

Flour Milling and Mill Engineering: Paper by B. W. Dedrick, instructor in mill engineering, Pennsylvania State College. Published in abstract, together with an abstract of the discussion, in The Journal for July 1912.

SAN FRANCISCO, APRIL 3

The Design and Mechanical Features of the California Gold

Dredge: Paper by Robert E. Cranston. Published in this volume.

NEW YORK, APRIL 9

Meeting in honor of the German Museum Commission, with an address by Dr. Oskar von Miller, President of the Verein deutscher Ingenieure, on the work of the Museum. A more complete account appears in The Journal for May 1912.

ST. LOUIS, APRIL 13

Address by Dr. Rudolph Diesel of Munich on the Development of the Diesel Engine.

NEW YORK, APRIL 30

Conferring of Honorary Membership upon Dr. Rudolph Diesel of Munich and address on the Development of the Diesel Engine, by Dr. Diesel. This address appears in The Journal for June 1912.

NEW HAVEN, APRIL 17

Meeting on Manufacturing Costs: Papers, The Development of Manufacturing Costs, Prof. J. W. Roe; Manufacturing Costs, by Bruce Fenn; Cost Department and its Relation to the Management, by G. P. Miller, secretary Bridgeport Brass Company. The latter paper was presented at the evening session.

An account of the meeting appears in The Journal for March 1912.

NEW YORK, MAY 14

Commercial Dictating Machines: Topical discussion by A. J. McFaul, of the Allen Advertising Company, New York, Otto Brushaber, Dictaphone Company, New York, and C. K. Fankhauser, American Telegraphone Company, Springfield, Mass. An account of the meeting appears in The Journal for August 1912.

BOSTON, MAY 17

Papers: Progress in Development of a New Type of Centrifugal Pump and Blower, by C. V. Kerr and A. L. Schaller; and Increase of Bore of High-Speed Wheels by Centrifugal Stresses, by Sanford A. Moss. The latter appears in this volume.

THE SPRING MEETING

The Spring Meeting was held at Cleveland, Ohio, May 28 to 31, 1912, with a total registration of 562, of whom 221 were members. The headquarters during the meeting were at the Chamber

of Commerce Building, where the professional sessions and several of the social functions were held, and lunches were served there on Wednesday and Friday. This building is well adapted for a gathering of this kind, and especially so in this case, as it is the home of the Cleveland Engineering Society, whose rooms were freely thrown open to the use of the members.

The Committee on Meetings had arranged a strong professional program, and the Local Committee contributed in every possible way to the pleasure of the members and their guests. The program follows:

PROGRAM

Tuesday Afternoon, May 28

Informal reception at the home of Mr. and Mrs. Swasey.

Tuesday Evening

Membership reunion and informal reception, Chamber of Commerce Hall.

Wednesday Morning, May 29

Business and Professonal Session

Business Meeting: Reports of committees, tellers of election; new business.

A New Analysis of the Cylinder Performance of Reciprocating Engines, J. Paul Clayton.

Discussed by Arthur L. Rice, F. E. Cardullo, R. C. Stevens, S. A. Moss, R. C. H. Heck, F. W. Marquis, C. D. Young, J. B. Stanwood, W. D. Ennis.

Equipment of a Modern Flour Mill on a Gradual Reduction System, John F. Harrison and W. W. Nichols.

Design and Mechanical Features of the California Gold Dredge, Robert E. Cranston.

Simultaneous Session

GAS POWER SECTION

Problems in Natural Gas Engineering, Thomas R. Weymouth.

Discussed by C. N. Cross.

Bituminous Coal Producers for Power, C. M. Garland.

Discussed by O. P. Hood, E. D. Dreyfus, T. A. Marsh, Chas. W. Baker.

Some Tests on Carbureters, George W. Munro.

Discussed by F. S. King, F. H. Vose, George A. Orrok.

Wednesday Afternoon

Inspection by members of local manufacturing and power plants. Automobile trip for ladies through the parks. Tea served at the Country Club.

Wednesday Evening

LECTURE

SOUND WAVES: HOW TO PHOTOGRAPH THEM AND WHAT THEY MEAN, Dr. Dayton C. Miller, of the Case School of Applied Science.

Thursday Morning, May 30

PROFESSIONAL SESSION

NEW PROCESSES FOR CHILLING CAST IRON, Thos. D. West.

Discussed by H. M. Howe, P. Munnoch, Paul Kreuzpointner, J. A. Beckett, C. Hering, A. S. Dowler, A. Sauveur, B. Stoughton, J. E. Johnson, Jr., H. M. Lane.

TESTS OF CHILLABLE IRONS, Thos. D. West.

Discussed by Henry Souther, Paul Kreuzpointner, C. B. Murray, B. Stoughton, A. E. Outerbridge, Jr.

STRENGTH OF STEEL TUBES, PIPES AND CYLINDERS UNDER INTERNAL FLUID PRESSURE, Reid T. Stewart.

Discussed by C. M. Sames, T. A. Marsh, S. A. Moss.

ON THE CONTROL OF SURGES IN WATER CONDUITS, Wm. F. Durand.

SPEED REGULATION IN HYDROELECTRIC PLANTS, Wm. F. Uhl.

Thursday Afternoon

Excursion on Lake Erie.

Thursday Evening

Reception and dance at Colonial Club.

Friday Morning, May 31

PROFESSIONAL SESSION

THE PRESENT STATE OF DEVELOPMENT OF LARGE STEAM TURBINES, A. G. Christie.

Discussed by George A. Orrok, F. Hodgkinson, C. V. Kerr, J. A. Moyer, Clarence P. Crissey, E. D. Dreyfus, Carl Geo. deLaval, W. L. R. Emmet.

A DISCUSSION OF CERTAIN THERMAL PROPERTIES OF STEAM, G. A. Goodenough.

Discussed by F. E. Cardullo, H. N. Davis, R. C. H. Heck, S. A. Moss.

THE REDUCTION IN TEMPERATURE OF CONDENSING WATER

Reservoirs due to Cooling Effect of Air and Evaporation, W. B. Ruggles.

Discussed by F. E. Cardullo, W. H. Carrier, W. T. Donnelly.

Results of Tests on the Discharge Capacity of Safety Valves, E. F. Miller and A. B. Carhart.

Discussed by C. H. Chase, W. Trinks, A. B. Carhart.

Friday Afternoon

Inspection of plants of The Goodrich Rubber Company, the Wellman-Seaver-Morgan Company at Akron, and the Diamond Match Company at Barberton.

No. 1347

A NEW ANALYSIS OF THE CYLINDER PERFORMANCE OF RECIPROCATING ENGINES

By J. Paul Clayton, Urbana, Ill.

Junior Member of the Society

The investigation described in this paper is the result of an extensive analytical and experimental study of the forms of the expansion and compression curves which occur in indicator diagrams. From this study has been developed a new and very complete analysis of cylinder performance for reciprocating engines using any elastic fluid for the working medium.

2 The analytical study was carried on by transferring the indicator diagram to logarithmic cross-section paper and thus drawing a figure which will be called a logarithmic diagram. It is well known that the equation of the polytropic curve $PV^n = C$ becomes a straight line when plotted on logarithmic cross-section paper. Conversely, when the expansion or compression curve of an indicator diagram becomes a straight line in the logarithmic diagram, then the curve is of the form $PV^n = C$, the value of n being the slope of the line.

3 The logarithmic diagram is more useful for analysis than any other form of diagram because of the natural limitations of the human mind. We do not possess the power to distinguish between curves, but are able, however, to see clearly the differences in these curves after they have been transformed into straight lines, which fact alone makes these new methods of analysis possible. We are now enabled to comprehend in their straight-line form curves which we have always seen, but could not distinguish one from the other in their original form.

4 By means of the logarithmic diagram it has been found that, free from certain abnormal influences, expansion or compression of an elastic medium takes place in the cylinders of reciprocating engines substantially according to the law $PV^n = C$.

Presented at the Spring Meeting, Cleveland 1912, of The American Society of Mechanical Engineers.

5 From the fact that the law $PV^n = C$ holds for expansion and compression curves from practice, there have been developed rational methods of approximating the clearance volume, of locating the cyclic events closely, and of detecting moderate leakage when the engine is in regular operation. These methods apply, however, only to those indicator diagrams which are taken from the cylinders of reciprocating engines using any elastic fluid for the working medium and having as a part of the cycle an expansion or compression of the medium.

6 It has been discovered that the value of n for the expansion curves of steam diagrams bears a definite relation in any given cylinder to the proportion of the total weight of steam mixture which was present as steam at cut-off. This proportion or quality will be called X_c in this investigation, and its value will be expressed in decimal parts of unity. The relation of the value of n to the value of X_c for the same class of cylinder as regards jacketing has been found to be practically independent of engine speed and of cylinder size.

7 The practical significance of finding this relation is that there is now available an accurate method of approximating the value of X_c and therefore the actual weight of steam and water present at cut-off from the indicator diagram alone.

8 It is believed, after working with the new methods of analysis, that the existing methods are in a very crude state, and that many of the deductions made from them are without foundation or meaning.

9 Preliminary to the investigation, an examination was made of the nature and form of the expansion and compression curves from a large number of steam engines, by the aid of the logarithmic diagram.

10 Values of n in the equation expressing the law $PV^n = C$, however, exhibited a large range of variation, the range being from 0.70 to 1.34. The engines from which the values were obtained differed in type, size, speed, steam pressure, ratio of expansion and back pressure. Obviously, comparisons could not be made of these examples because of the number and magnitude of the variables.

11 Indicator diagrams taken from the same cylinder with different cut-off positions showed that the value of n was higher as the cut-off was lengthened. There was a large variation in the value of n where the conditions of cylinder size, speed, steam pressure and steam distribution were the same. The only variable shown by the diagrams was length of the cut-off. After considerable study, it was

decided that there might be some relation in any one cylinder between the value of n for the expansion curve and the value of X_c, as the value of X_c was known to be higher as the length of cut-off increased. One fact that seemed to confirm this hypothesis was that, with superheated steam (under the same general conditions, except the kind of steam used), when the value of X_c is high, the value of n for expansion is always much higher than with saturated steam. The only important variable between the use of saturated and superheated steam to account for the change in the value of n was the quality of the steam mixture at cut-off, or the value of X_c.

12 All cases of engines examined using superheated steam at normal cut-off showed n to be higher than 1.0, and as high as 1.34; and all cases of small engines using saturated steam showed n to be lower than 1.0 and as low as 0.70. These facts led to the conclusion that the value of X_c was the most important single factor in the accompanying value of n. Tests were therefore planned in which the effort was made to vary the value of X_c between the widest practicable limits.

LABORATORY TESTS

13 An outline of these tests will first be given, with a synopsis of the method of analysis used, after which the application of the analysis to indicator diagrams will be given for the purpose of determining cylinder performance.

14 A single-cylinder, long-range cut-off, 12-in. by 24-in. Corliss engine, located in the mechanical engineering laboratory of the University of Illinois, was selected for the tests. A Corliss engine was selected because of the fact that in this type all the steam used passes through the cylinder.

15 It was planned to observe the effect upon the value of n of varying the value of X_c under different conditions of pressure and speed; n being the exponent in the equation $PV^n=C$ for the expansion curve of an indicator diagram, and X_c the quality of the steam at cut-off. The value of X_c was varied through a large range by the use of saturated and superheated steam, in conjunction with different lengths of cut-off under the same conditions of pressure and speed. The values of X_c obtained ranged from 0.50 to 0.90, covering the range usually found in practice with the type of engine used.

16 The values of n for the expansion curves were obtained by means of the logarithmic diagram explained in detail in the Appendix. The value of X_c given in the log is the average of the results

obtained from one set of head-end and crank-end diagrams for each test. The unit of measurement, therefore, was the revolution, as the values of X_e for the head and crank ends cannot be measured separately when one exhaust pipe is used for both ends of the cylinder. The value of n given for one test is the average of the separate values from the expansion curves of the head-end and crank-end diagrams, taken from the set of diagrams already mentioned.

17 Seventy-four tests in 16 series were run. Of this number, 60 tests in 14 series were selected as fulfilling the requirements decided upon to give reliable data. Each series consisted of four to five separate tests, differing from each other only in the length of cut-off, with the same conditions of pressure and speed. All tests were run with the steam exhausting from the cylinder at about atmospheric pressure into a surface condenser. The length of cut-off was varied in nearly uniform steps from about 5 per cent to 45 per cent of the length of the stroke, and was the means of varying the value of X_e when using either saturated or superheated steam.

18 The 14 series were divided into two divisions of seven series each; one division being run with saturated steam, and the other with steam superheated to 500 deg. fahr. at the superheater. Each division consisted of five series run at different gage pressures at constant speed, and two series run at different speeds with constant pressure. The steam pressures used were 57.5, 76.5, 95, 113, and 132 lb. gage with the engine running at 120 r.p.m. The other speeds employed were 90 r.p.m. and 150 r.p.m. at the gage pressure of 113 lb. Each division, therefore, gave the effect of the use of five steam pressures at constant speed, and three speeds at constant pressure.

19 The governor change-speed device was always set to give the desired speed with the engine running at no load. As the load was increased, the speed decreased through the action of the governor in about the same proportion for all initial speeds. Whenever speed is mentioned, the no-load speed is the one referred to, the exact speed for any one test being given in the general log.

20 The general log of the 29 tests run with saturated steam is given in Table 1. Table 2 contains the results of the 31 tests run with superheated steam. Table 3 gives the averages of similar series, called groups, run at the same pressure and speed, with both saturated and superheated steam. For any one group of the two series of tests, as has already been pointed out, the only variables are the length of cut-off and the value of X_e.

TABLE 1 GENERAL LOG. TESTS WITH SATURATED STEAM

Series	Number of Test	Pressure Lb. Per Sq. Inch				Revolutions per Hour	Weight of Condensate per Hour, lb.	Weight of Condensate per Revolution, lb.	Lb. of Steam per Revolution shown by Indicator at Cut-Off	Lb. of Compression saved per Revolution by Indicator	Lb. of Steam Mixture Actually present per Revolution	Quality of Steam Mixture at Cut-Off, Parts of Unity	Absolute Pressure at Cut-Off, Lb.	Average Value of n from Expansion Curves	Cut-Off, Per Cent of Stroke	Average Absolute Pressure at Cut-Off for Series, Lb
		At Throttle by Gage	Barometer	Absolute Pressure at Throttle	Average Absolute Pressure of Series at Throttle											
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
I	2	57.8	14.3	72.1	....	6428	1054	0.1640	0.0929	0.0248	0.1888	0.492	60.3	0.926	14.6	....
	3	57.3	14.2	71.5	71.3	6100	1104.5	0.1811	0.1037	0.0221	0.2032	0.510	61.3	0.882	16.5	60.6
	4	55.8	14.5	70.3	....	6302	2010	0.3189	0.2099	0.0276	0.3465	0.606	60.2	0.961	41.2	....
II	6	77.0	14.3	91.3	....	6846	1010	0.1475	0.0846	0.0235	0.1710	0.495	76.8	0.927	8.1	....
	7	76.7	14.2	90.9	90.5	6440	1384	0.2149	0.1281	0.0230	0.2379	0.539	80.2	0.895	15.7	77.9
	8	75.6	14.5	90.1	....	6640	2104	0.3168	0.2089	0.0253	0.3421	0.610	77.4	0.949	30.9	....
	9	75.0	14.5	89.5	....	6282	2559	0.4073	0.2727	0.0260	0.4333	0.629	77.4	0.950	42.6	....
IV	15	95.8	14.3	110.1	....	6940	1048	0.1510	0.0903	0.0236	0.1746	0.517	95.6	0.910	6.1	....
	16	96.0	14.2	110.2	....	6510	1530	0.2350	0.1462	0.0226	0.2756	0.531	95.6	0.887	14.6	....
	17	94.9	14.2	109.1	109.1	6432	1584	0.2462	0.1533	0.0233	0.2695	0.569	96.2	0.914	15.5	95.3
	18	94.5	14.4	108.9	.....	6750	2298	0.3404	0.2145	0.0246	0.3650	0.588	92.5	0.996	25.4	....
	19	93.6	14.4	108.0	....	6410	2758	0.4302	0.2952	0.0263	0.4565	0.647	95.6	1.002	37.0	....
	20	94.0	14.4	108.4	....	5958	3008	0.5048	0.3484	0.0248	0.5296	0.658	96.1	0.963	44.3	....
V	22	114.6	14.3	128.9	....	7046	1030	0.1461	0.0907	0.0239	0.1700	0.533	105.5	0.835	4.8	....
	23	113.7	14.2	127.9	127.7	6688	1566	0.2341	0.1445	0.0240	0.2581	0.560	110.0	0.944	11.3	109.4
	24	113.6	14.2	127.8	....	6466	1738	0.2687	0.1776	0.0227	0.2914	0.610	109.1	0.940	15.7	....
	25	112.0	14.2	126.2	....	6736	2664	0.3954	0.2708	0.0253	0.4207	0.644	113.0	1.004	27.3	....
VI	27	131.9	14.3	146.2	....	7086	986	0.1391	0.0941	0.0240	0.1631	0.576	126.7	0.946	3.7	....
	28	133.2	14.2	147.4	....	6862	1466	0.2136	0.1426	0.0231	0.2367	0.602	130.8	0.950	8.3	....
	29	132.0	14.2	146.2	146.2	6306	2000	0.3172	0.2050	0.0220	0.3392	0.604	130.0	1.006	16.4	129.4
	30	130.8	14.2	145.0	....	6840	2796	0.4087	0.2760	0.0256	0.4343	0.635	128.9	1.008	24.9	....
	31	132.0	14.4	146.4	....	6138	3276	0.5337	0.3762	0.0269	0.5606	0.671	130.6	1.010	35.3	....
VII	32	113.9	14.4	128.3	....	5258	1025	0.1949	0.0909	0.0251	0.2200	0.413	112.1	0.854	4.9	....
	33	113.9	14.4	128.3	128.4	5102	1585	0.3106	0.1641	0.0242	0.3348	0.490	113.2	0.932	12.8	113.7
	34	114.1	14.4	128.5	....	4940	2090	0.4230	0.2402	0.0246	0.4476	0.537	115.7	0.930	23.2	....
VIII	36	113.3	14.4	127.7	....	8772	1616	0.1842	0.1136	0.0260	0.2102	0.540	108.4	0.927	7.7	....
	37	114.2	14.4	128.6	127.9	8518	2554	0.2998	0.1990	0.0254	0.3252	0.612	110.3	0.964	19.7	110.3
	38	112.8	14.4	127.2	....	8182.5	3520	0.4302	0.3080	0.0256	0.4558	0.676	110.1	1.045	34.4	....
	39	113.5	14.4	127.9	....	7625	4142.5	0.5433	0.3968	0.0259	0.5692	0.697	112.2	1.016	46.0	....

TABLE 2 GENERAL LOG. TESTS WITH SUPERHEATED STEAM

Series	Number of Test	Pressure Lb. Per Sq. Inch: At Throttle by Gage	Pressure Lb. Per Sq. Inch: Barometer	Pressure Lb. Per Sq. Inch: Absolute Pressure at Throttle	Pressure Lb. Per Sq. Inch: Average Absolute Pressure of Series at Throttle	Revolutions per Hour	Weight of Condensate per Hour, Lb.	Weight of Condensate per Revolution, Lb.	Lb. of Steam per Revolution shown by Indicator at Cut-off	Lb. of Compression saved per Revolution by Indicator	Lb. of Steam Mixture Actually present per Revolution	Quality of Steam Mixture at Cut-Off, Parts of Unity	Absolute Pressure at Cut-Off, Lb.	Average Value of n from Expansion Curves	Cut-Off, Per Cent of Stroke	Average Absolute Pressure at Cut-Off for Series, Lb.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
IX	40	59.5	14.5	74.0	....	7040	815	0.1158	0.0838	0.0232	0.1390	0.602	61.2	0.904	11.7	....
	41	60.5	14.5	75.0	73.4	6856	1058	0.1543	0.1235	0.0239	0.1782	0.693	61.6	1.053	21.6	61.1
	42	57.7	14.5	72.2	....	6522	1288	0.1975	0.1738	0.0228	0.2203	0.788	60.6	1.114	32.0	....
	43	57.8	14.5	72.3	....	6262	1396	0.2229	0.2110	0.0227	0.2456	0.859	60.8	1.144	41.3	....
X	44	76.1	14.5	90.6	....	6655	1201.7	0.1806	0.1106	0.0246	0.2052	0.539	79.2	0.951	12.4	....
	45	77.4	14.5	91.9	91.0	6550	1294	0.1976	0.1342	0.0250	0.2226	0.603	79.2	0.957	16.5	78.5
	46	76.3	14.5	90.8	....	6290	1455	0.2313	0.1910	0.0230	0.2543	0.751	77.4	1.105	27.6	....
	47	76.0	14.5	90.5	....	5908	1822	0.3084	0.2720	0.0226	0.3310	0.822	78.0	1.112	41.5	....
XI	48	94.3	14.5	108.8	....	7112	822	0.1156	0.0877	0.0228	0.1384	0.634	92.8	0.924	6.1	....
	49	93.0	14.5	107.5	....	6936	1270	0.1831	0.1311	0.0234	0.2065	0.635	90.7	0.982	13.3	....
	50	95.5	14.5	110.0	109.0	6682	1660	0.2484	0.2040	0.0226	0.2710	0.753	97.1	1.040	22.3	94.1
	51	94.7	14.4	109.1	....	6500	1780	0.2738	0.2479	0.0220	0.2958	0.838	96.4	1.116	28.8	....
	52	95.1	14.4	109.5	....	6240	2072	0.3321	0.2821	0.0226	0.3547	0.796	93.6	1.001	39.2	....
XII	53	115.2	14.4	129.6	....	7100	914	0.1287	0.0972	0.0224	0.1511	0.642	111.5	0.991	5.1	....
	54	114.3	14.4	128.7	....	6932	1380	0.1991	0.1561	0.0225	0.2216	0.705	109.0	0.990	13.0	....
	55	114.1	14.4	128.5	128.4	6662	1906	0.2861	0.2419	0.0226	0.3087	0.783	111.7	1.032	22.9	111.0
	56	114.1	14.3	128.4	....	6326	2315	0.3660	0.3230	0.0219	0.3879	0.833	111.5	1.140	35.1	....
	57	112.3	14.5	126.8	....	5940	2350	0.3956	0.3709	0.0228	0.4184	0.886	111.5	1.209	40.3	....
XIV	61	132.5	14.3	146.8	....	7136	922	0.1292	0.0935	0.0228	0.1520	0.615	127.7	0.939	3.2	....
	62	128.0	14.3	142.3	....	7010	1384	0.1974	0.1553	0.0229	0.2203	0.715	123.6	0.969	11.0	....
	63	131.3	14.3	145.6	145.4	6810	1789	0.2627	0.2331	0.0228	0.2855	0.817	131.0	1.074	18.1	128.5
	64	131.0	14.3	145.3	....	6602	2171	0.3288	0.3113	0.0222	0.3510	0.887	130.5	1.190	26.9	....
	65	132.6	14.5	147.1	....	6088	2640	0.4336	0.3988	0.0230	0.4566	0.873	129.5	1.165	37.1	....
XV	66	114.8	14.5	129.3	....	5322	808	0.1518	0.0950	0.0220	0.1738	0.546	112.3	0.948	4.6	....
	67	114.2	14.5	128.7	128.0	5212	1218	0.2337	0.1540	0.0222	0.2559	0.602	111.1	0.941	12.4	111.7
	68	112.3	14.5	126.8	....	5018	1574	0.3137	0.2356	0.0226	0.3363	0.700	110.7	1.013	23.1	....
	69	112.7	14.5	127.2	....	4817	1809	0.3755	0.3078	0.0217	0.3972	0.774	112.5	1.148	32.0	....
XVI	71	112.8	14.5	127.3	....	8700	1512	0.1738	0.1447	0.0249	0.1987	0.729	108.0	1.093	12.3	....
	72	111.8	14.5	126.3	126.9	8438	2018	0.2392	0.2080	0.0257	0.2649	0.786	109.0	1.091	20.5	108.3
	73	112.2	14.5	126.7	....	8084	2458	0.3041	0.2970	0.0236	0.3277	0.907	107.2	1.213	32.4	....
	74	112.6	14.5	127.1	....	7494	3110	0.4150	0.4135	0.0237	0.4387	0.943	108.8	1.234	46.9	....

TABLE 3 LOG OF GROUP AVERAGES

Group	Series	Number of Tests Inclusive	Pressure, Lb. per Sq. In.			Absolute Pressure from Indicator Diagram, Lb. per Sq. In.					Specific Volume of Steam at Pressure Adopted	Number of Tests in Series	Number of Tests in Group	Number of Diagrams Analyzed	Average Quality of Steam Mixture at Cut-Off from Tests in Group, Parts of Unity	Average Value of n from Tests in Group
			Average Absolute Pressure at Throttle by Gage	Average Absolute Pressure of Group	Difference from Group Average of Next Lower Pressure	Average at Cut-Off	Average of Group	Difference from Group Average of Next Lower Pressure	Drop in Pressure, Throttle to Cut-Off	Pressure Adopted to find Quality of Steam Mixture at Cut-Off						
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
A	I	2–4	71.3	72.4	....	60.6	60.9	....	11.5	61.0	7.06	3	7	14	0.6500	0.998
	IX	40–43	73.4	....	....	61.1	....	....	....	....	....	4	....	....	....	....
B	II	6–9	90.5	90.8	18.4	77.9	78.2	17.3	12.6	78.0	5.60	4	8	16	0.6235	0.981
	X	44–47	91.0	....	....	78.5	....	....	....	....	....	4	....	....	....	....
C	IV	15–20	109.1	109.1	18.3	95.3	94.7	16.5	14.4	95.0	4.65	6	11	22	0.6515	0.976
	XI	48–52	109.	....	....	94.1	....	....	....	....	....	5	....	....	....	....
D	V	22–25	127.7	127.9	18.8	109.4	110.7	16.0	17.2	111.0	4.012	4	9	18	0.6885	1.010
E	VII	32–34	128.4	....	....	113.7	....	....	....	....	....	3	7	14	0.5803	0.967
F	VIII	36–39	127.9	....	....	110.3	....	....	....	....	....	4	8	16	0.7363	1.073
D	XII	53–57	128.4	....	....	111.0	....	....	....	....	....	5	....	....	....	....
E	XV	66–69	128.0	....	....	111.7	....	....	....	....	....	4	....	....	....	....
F	XVI	71–74	126.9	....	....	108.3	....	....	....	....	....	4	....	....	....	....
G	VI	27–31	146.2	145.8	17.9	129.4	129.0	18.3	16.8	129.0	3.478	5	10	20	0.6995	1.026
	XIV	61–65	145.4	....	....	128.5	....	....	....	....	....	5	....	....	....	....

21 *Values of n obtained under Different Conditions from the Same Engine Cylinder.* All the simultaneous values of X_c and n obtained from the 60 tests were plotted in Fig. 1. A study of this figure shows beyond question that as X_c increases in value, n increases also. The values all lie in a region which has a definite trend towards higher simultaneous values of X_c and n. Observing the general trend, it is seen that there is no exception to this general relation. No value of n below 1.00, for instance, is found for values of X_c above 0.80, and no value of n above 1.10 is found for values of X_c below 0.72.

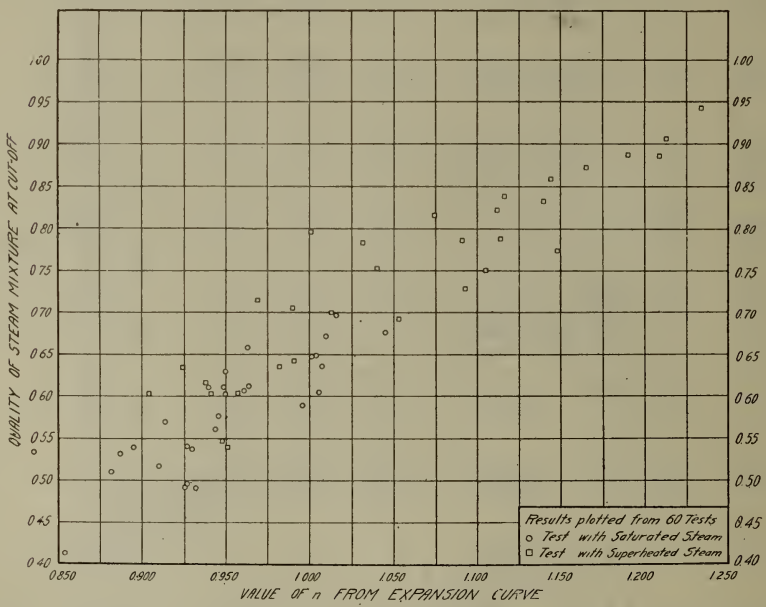


FIG. 1 GENERAL RELATIONS BETWEEN QUALITY AT CUT-OFF AND THE VALUE OF n FOR VARIOUS PRESSURES AND SPEEDS

22 It is also apparent that the points with long cut-off, obtained from saturated steam for given values of X_c and n, are in the same region occupied by the points with short cut-off obtained from superheated steam for the same given values of X_c and n. Examining the region of $n = 0.90 - 1.00$ and of $X_c = 0.50 - 0.70$, we see that the points obtained with long cut-off with saturated steam, and with short cut-off with superheated steam, lie indiscriminately together. This shows conclusively, in a general way, that the value of n is practically independent of the length of cut-off, even though this

length may vary from 5 per cent to 45 per cent, and that n depends solely on the value of X_o, the only other variable.

23 The points shown in Fig. 1 occupy a relatively wide region until they are separated into the various groups of similar pressures and speeds.

24 The points for each group were plotted separately, and separate curves were determined for each condition. From preliminary plotting, the relations between n and X_o were found to be expressed closer by straight lines than by any other family of curves. The

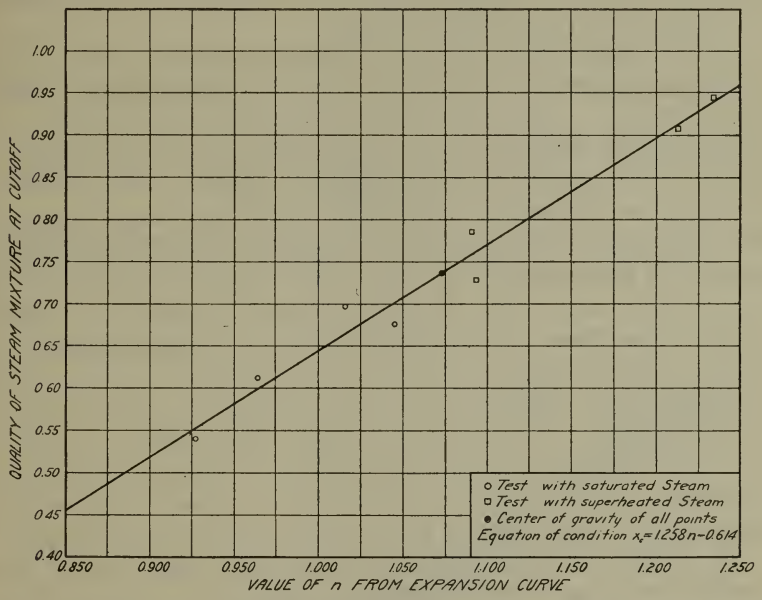


Fig. 2 Relation of Quality and the Value of n for Tests run at 111 Lb. Absolute Pressure of Cut-Off and 150 R.P.M.

method used to draw these lines will be given for group F, which includes series VIII, with four tests, and series XVI with four tests. This group is shown in Fig. 2. All points were given equal weight. The average of all the coördinates, or the "center of gravity" was found and the conditions imposed that the line pass through this center as an axis, and that the slope be determined by the position of the points. The points in group F were divided into four logical pairs or groupings, and the center of gravity found for each grouping. The line was then drawn as shown. Where points were located

so that a logical grouping was in doubt, various groupings were made, and each given weight in determining the slope.

25 The equation of the curve selected is

$$X_e = 1.258\, n - 0.614$$

The average deviation of the points from this line is 2.6 per cent (measured from the zero of X_e) and the maximum deviation is 4.6 per cent. This average deviation, 2.6 per cent, is smaller than that for most of the groups. The straight line, in most cases, represents the points found as close as any other curve that could be employed, and has the merit of simplifying greatly the subsequent use made of the relations for the different groups.

26 The values of n for group F were also plotted to the various accompanying cut-off positions at which each test was run. This is shown in Fig. 3. The points in Fig. 3 show that for each initial condition of steam the value of n increased as the cut-off was lengthened, and that with any given cut-off different values of n were obtained according as saturated or superheated steam was used. Thus at a cut-off of 15 per cent, the value of n obtained is 0.950 with saturated steam, and 1.084 with superheated steam. The only variable present in these two cases is X_e, the value of which is higher with superheated steam than with saturated steam. A definite relation between n and the length of cut-off occurs only when a given cut-off is accompanied by the same value of X_e, that is, the value of n bears a direct relation to X_e but not to cut-off. Fig. 3 taken in conjunction with Fig. 2 proves that the value of n depends directly only upon the value of X_e, and that the relation of n and X_e is practically independent of the length of cut-off within the limits of the tests.

27 *Effect of Varying the Steam Pressure at Constant Speed.* The relations of X_e and n were determined separately for all groups by the method outlined for Fig. 2. The lines for the five pressures used, comprising the results of groups A, B, C, D and G, were replotted as shown in Fig. 4. This figure also contains other curves that are discussed in Paragraph 38. The lines shown give the relations of X_e and n for various absolute pressures at cut-off, all obtained with a speed of 120 r.p.m.

28 These curves were then examined to find the effect of varying the absolute pressure at cut-off (designated as p) on the relations of X_e and n. In Fig. 4, the constant pressure curves were intercepted at constant values of n, and the coördinates of X_e and p for the points of intersection plotted in Fig. 5. This process was repeated at intervals of 0.05 for the values of n from 0.850 to 1.250. The

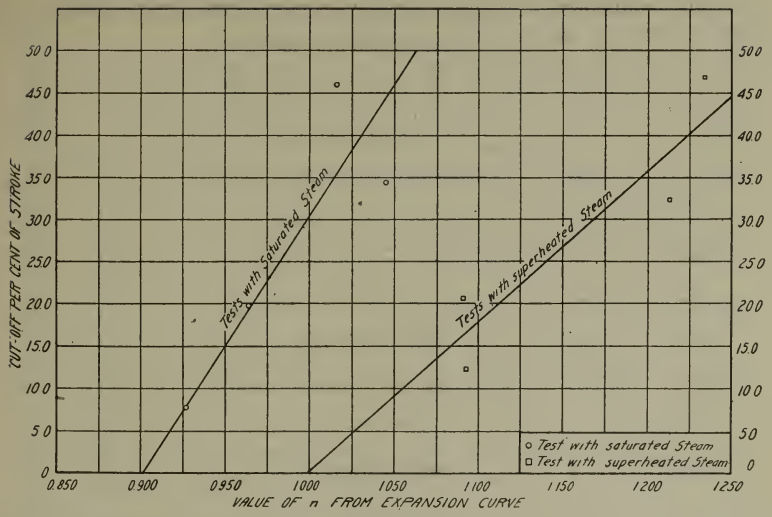


FIG. 3 RELATIONS OF PER CENT OF CUT-OFF AND THE VALUE OF n FOR TESTS RUN AT 111 LB. ABSOLUTE PRESSURE AT CUT-OFF AND 150 R.P.M.

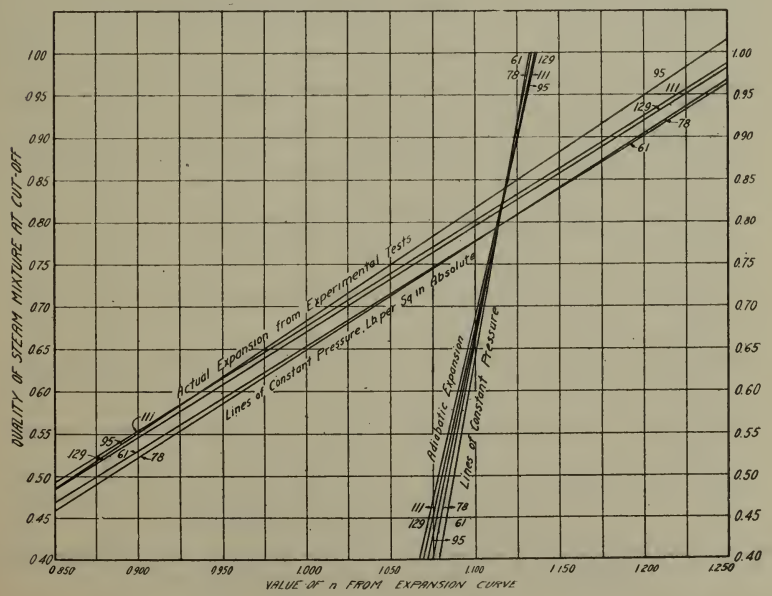


FIG. 4 RELATIONS AT CUT-OFF, BETWEEN QUALITY AND THE VALUE OF n FOR VARIOUS PRESSURES AT CONSTANT SPEED

resulting points did not give smooth curves, but, after considerable study and trial, curves were adopted as shown.

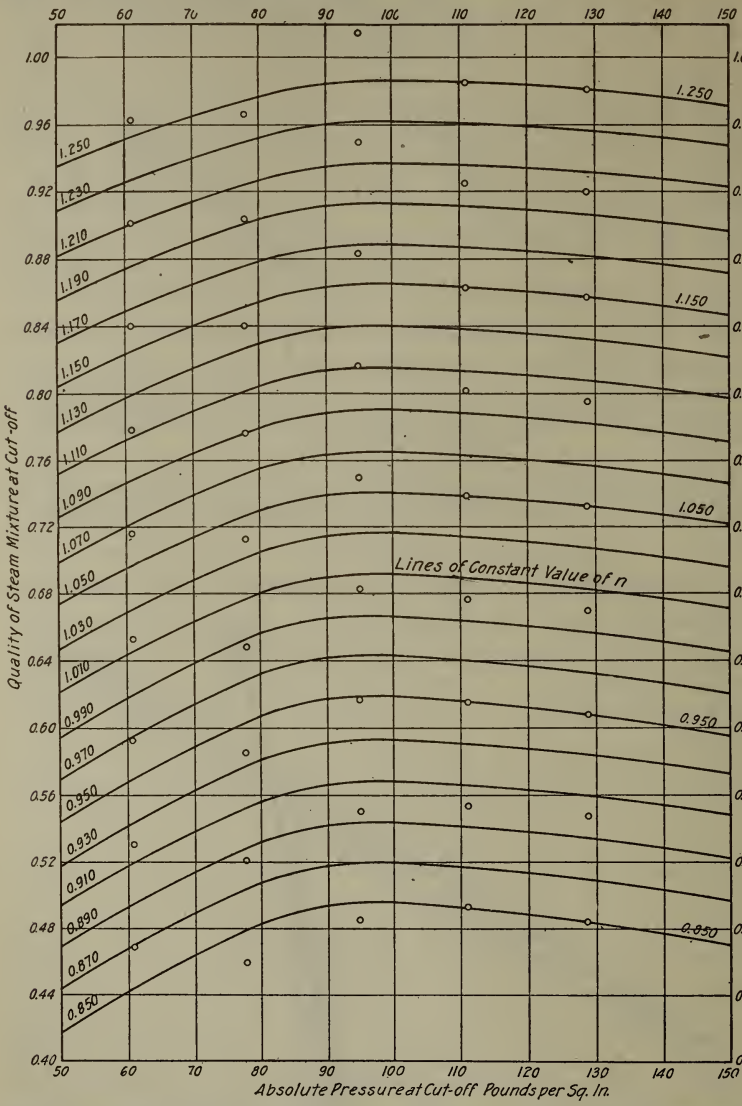


FIG. 5 THE RELATION AT CUT-OFF, BETWEEN QUALITY AND PRESSURE FOR CONSTANT VALUES OF n FROM EXPANSION CURVE

29 This procedure gave a series of relations between X_c and p for constant values of n. Since, however, the independent variables

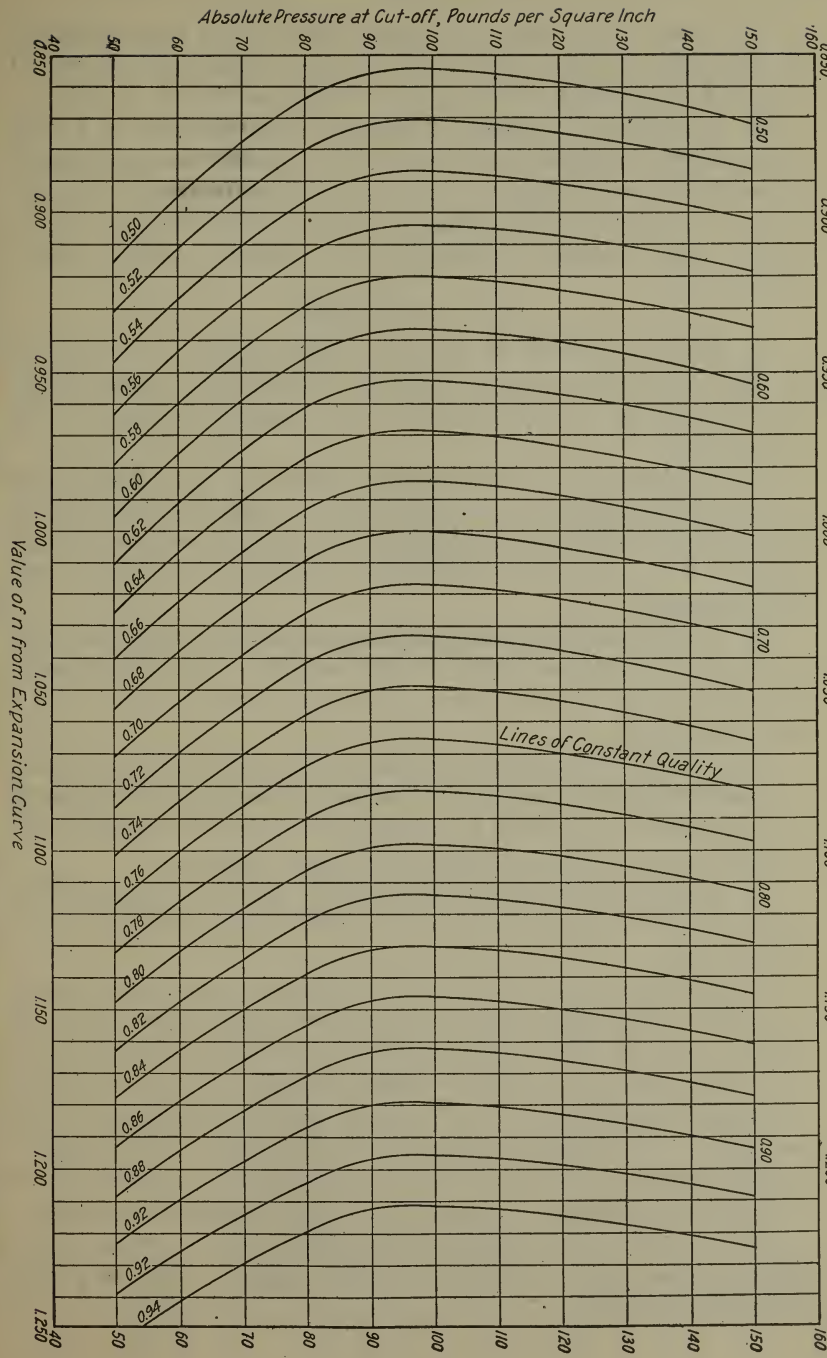


FIG. 6 THE RELATION BETWEEN PRESSURE AT CUT-OFF AND VALUES OF n FOR CONSTANT QUALITY AT CUT-OFF

in any actual curve under examination are n and p, the coördinates of the curves of Fig. 5 were changed so as to show the relations of n and p at constant values of X_c. These are shown in Fig. 6.

30 The effect of a change of pressure on the relations of X_c and n is not great between the limits of 75 to 150 lb. An approximate equation has been worked out, therefore, which represents the relations of X_c and n at an average pressure between the limits mentioned. The equation corresponds to the relations at 129 lb. and is of the form

$$X_c = 1.245\ n - 0.576$$

31 *Effect of Varying the Speed at Constant Pressure.* Groups D, E, and F were run at speeds of 120, 90, and 150 r.p.m., respectively, with the average cut-off pressure on the diagrams constant at 111 lb. absolute.

32 The regulation of the governor was very poor, there being about a 10 per cent drop in speed from no load to full load. For this reason the relations of X_c and n with various speeds at constant pressure were affected by considerable variation of the speed itself for each group.

33 The relations for each group were found as already described, and the curves plotted in Fig. 7. The relation of speed (designated as s) and X_c for constant values of n was derived from Fig. 7, and is given in Fig. 8.

34 The apparent relations of X_c, s, and n, obtained by drawing a smooth curve through the three points obtained for each value of n, is not satisfactory owing to insufficient data and the change of the speed itself in the three groups due to poor regulation.

35 The drop in the speed, for one group, does not seriously affect, however, the relations of X_c and n for the various pressures at constant speed.

36 *Relation of the Value of n to the Quality of the Steam Mixture at Cut-Off.* From the evidence obtained from these tests, it may be stated positively that for any one engine running at a given pressure and speed, there is a definite relation between X_c and n which is practically independent of the cut-off position within the limits examined. This relation is apparently a linear one. It may also be stated that the relation of X_c and n is dependent, to some extent, on the absolute pressure at cut-off, and on the speed of the engine.

37 It remained to compare the relations of X_c and n for the engine tested with the relations for other engines. This comparison is made in Table 4.

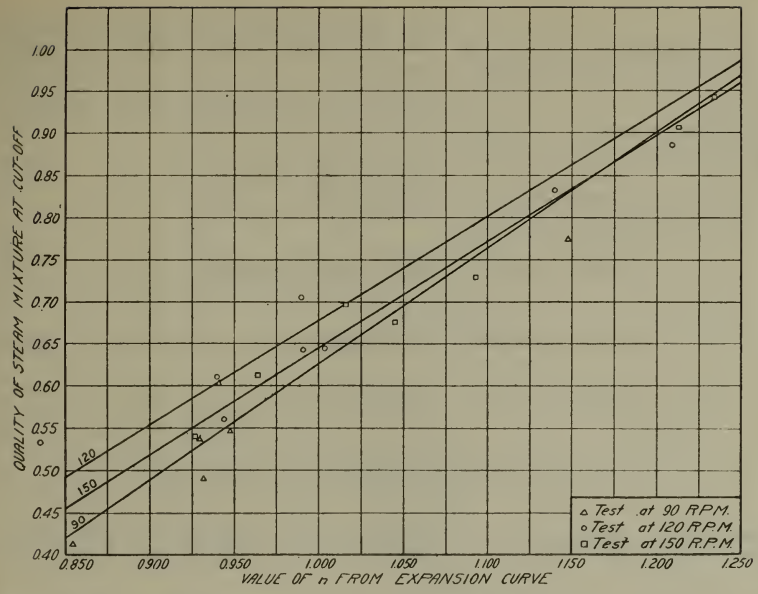


FIG. 7 RELATION OF QUALITY AT CUT-OFF AND THE VALUES OF n FOR VARIOUS SPEEDS AT THE CONSTANT CUT-OFF PRESSURE OF 111 LB.

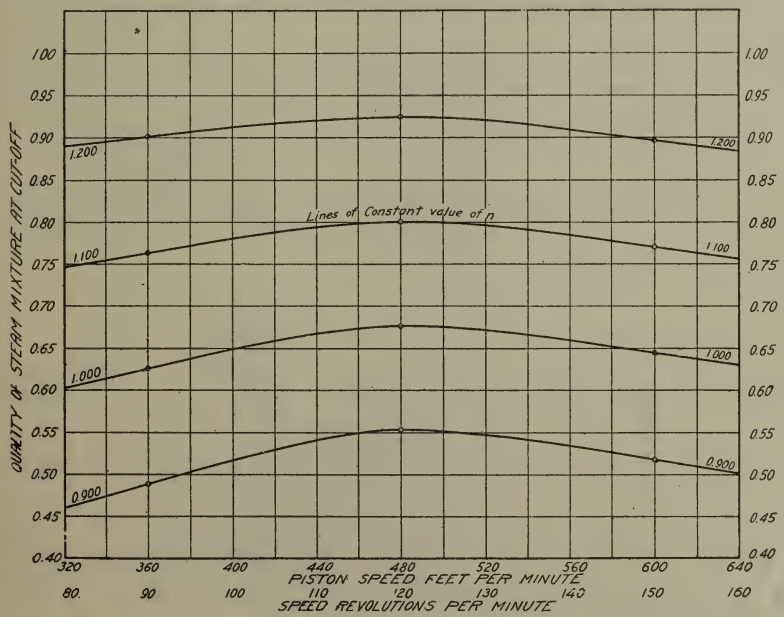


FIG. 8 RELATIONS, AT CUT-OFF, BETWEEN QUALITY AND SPEED FOR CONSTANT VALUES OF n AT CONSTANT PRESSURE

TABLE 4 APPROXIMATION OF THE ACTUAL STEAM CONSUMPTION FROM INDICATOR DIAGRAMS TAKEN FROM NON-JACKETED STEAM ENGINES

Number of Analysis	Class of Engine	Cylinder Dimensions, In.	Piston Rod Diameter, In.	Clearance, Per Cent of Piston Displacement	Actual Weight of Steam Passing through Cylinder, Lb. per Hour	Total I.H.P. Developed	Actual Steam per I.H.P-Hr., Lb.	Revolutions per Minute	Revolutions per Hour	Weight of Steam Passing through Cylinder per Revolution, Lb.	Weight of Steam Shown by Indicator at Cut-Off, Lb.	Weight of Steam Retained in Compression per Revolution by Indicator, Lb.
1	2	3	4	5	6	7	8	9	10	11	12	13
	Simple Corliss, 2-Valve, and 4-Valve, Non-Condensing Engines											
101	Corliss, single cylinder	23 x60	3¼	2.5	8477	305.2	27.77	74.7	4482	1.890	1.666	0.130
102	Corliss, single cylinder	28½x59½	4	3.0	13070	506.5	25.80	64.8	3888	3.362	2.822	0.167
103	Corliss, single cylinder	26 $\frac{7}{8}$x48	3⅞	3.0	6742	232.3	29.03	64.7	3882	1.736	1.402	0.116
104	Double-valve, 2 cylinders	17 x24.2	2¾	2.0	7952	310.1	25.64	152.9	9174	0.8670	0.7290	0.0635
105	Four-valve, single cylinder	16$\frac{5}{8}$x32	2¾	4.0	1879	50.2	37.43	79.8	4788	0.3925	0.2732	0.0607
106*a*	Four-valve, single cylinder	10½x12	2	5.0	2353	97.9	24.04	263	15780	0.1491	0.1377	0.0242
106*b*	Same engine as 106*a*	.....	...	....	1878	79.2	23.71	258	15480	0.1213	0.0959	0.0160
106*c*	Same engine as 106*a*	.....	...	....	997	41.7	23.90	260	15600	0.0639	0.0537	0.0260
106*d*	Same engine as 106*a*	.....	...	....	693	25.8	26.87	263	15780	0.0439	0.0402	0.0351
107	Corliss, 2 cylinders	16 x42	2¾	2.5	8873	342.43	25.91	84.9	5094	1.742	1.420	0.041
108*a*	Four-valve, single cylinder	18 x30	2⅝	5.0	4707	213.2	22.08	165.6	9936	0.4735	0.3890	0.0330
108*b*	Same engine as 108*a* (condensing)	.....	...	....	6050.2	209.1	28.93	164.4	9864	0.6137	0.5657	0.0712
109	Corliss, 2 cylinders (1 condensing)	23 x60	3⅜	3.0	13187	615.1	21.44	61	3660	3.602	3.140	0.080
110	Corliss, 1 cylinder (1 end condensing)	34.2x60	4⅞	3.0	11613	523.43	22.19	53.3	3198	3.633	2.835	0.093
111	Corliss, 2 cylinders (1 condensing)	28 x60	4	2.5	14295.7	562.1	25.43	60.27	3616.2	3.953	2.860	0.250
	Compound Condensing Engines—H.P. Cylinders											
151	Corliss cross-compound	20 x36x36	3	2.5	6740	433.5	15.00	75.83	4550	1.481	1.244	0.1012
152	Buckeye cross-compound	18¾x36x36	3½	4.3	8675	639.2	14.20	123.3	7398	1.173	1.023	0.1490
153	Corliss cross-compound	15x30⅛x30	2¾	4.0	2007	140.78	14.26	47.98	2879	0.6970	0.5492	0.0218
	Triple-Expansion Condensing Engines—I. P. Cylinders											
201	Corliss (slow-speed pumping engine)	24⅛x34x54x36	4½	2.10	4622	328.05	14.09	27.66	1659.6	2.787	1.432	0.031
	High-Speed Single-Valve, Non-Condensing Engines											
401	Piston valve, single cylinder	16 x16	2⅝	13.15	4820	184.6	26.35	253.4	15204	0.3196	0.3060	0.0948
402	Single valve, single cylinder	14½x13	2¼	10.0	1740	53.26	32.67	246	14760	0.1179	0.1197	0.0613
403	Single valve, single cylinder	14½x13	2¼	12.0	2018.6	61.7	32.71	248.4	14904	0.1353	0.1314	0.0594
	Simple Locomotive Engines											
451	Two cylinder slide valve	16 x24	2¾	H C R 7.44 7.98 L 7.34 7.63	7180	252.79	28.40	97.08	5824.8	1.232	1.0164	0.1947

Number of Analysis.	Weight of Steam: Mixture Actually Present per Revolution, Lb.	Quality of Steam Mixture at Cut-Off (X_c), Parts of Unity	Average Value of n from Expansion Curves	Absolute Pressure at Cut-Off, Lb. per Sq. In.	Approximation from Chart: Value of X_c from Chart, Parts of Unity	Approximation from Chart: Per Cent that the Value of X_c from Chart is Higher (+) or Lower (−) than Test Results	Approximation from Chart: Total Weight of Steam Mixture Present per Revolution, Lb.	Approximation from Chart: Weight of Steam: Passing through Cylinder per Revolution, Lb.	Approximation from Chart: Weight of Steam: Passing through Cylinder per Hour, Lb.	Approximation from Chart: Weight of Steam: Per I.H.P-hr., Lb.	Approximation from Chart: Per Cent that Total Weight of Steam per Hour is Higher (+) or Lower (−) than Test Results	Approximation from Equation $X_c = 1.245n - 0.576$: Value of X_c from Equation $X_c = 1.245n - 0.576$	Approximation from Equation $X_c = 1.245n - 0.576$: Per Cent that the Value of X_c from Equation is Higher (+) or Lower (−) than Test Results	Approximation from Equation $X_c = 1.245n - 0.576$: Total Weight of Steam Mixture present per Revolution, Lb.	Approximation from Equation $X_c = 1.245n - 0.576$: Weight of Steam: Passing through Cylinder per Revolution, Lb.	Approximation from Equation $X_c = 1.245n - 0.576$: Weight of Steam: Passing through Cylinder per Hour, Lb.	Approximation from Equation $X_c = 1.245n - 0.576$: Weight of Steam: Per I.H.P-Hr., Lb.	Approximation from Equation $X_c = 1.245n - 0.576$: Per Cent that Total Weight of Steam per Hour is Higher (+) or Lower (−) than Test Results	Method Giving Closer Approximation to Test Results, Chart (*C*), Equation (*E*)
1	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32
Simple Corliss, 2-Valve, and 4-Valve, Non-Condensing Engines																			
101	2.020	0.825	1.098	65	0.767	−7.0	2.172	2.042	9160	30.00	+8.1	0.790	−4.2	2.110	1.980	8880	29.18	+4.8	*E*
102	3.529	0.801	1.108	75	0.795	−0.7	3.551	3.384	13150	25.96	+0.7	0.804	+0.4	3.512	3.345	13000	25.57	−0.5	*E*
103	1.852	0.757	1.051	75	0.732	−3.3	1.915	1.799	6985	30.06	+3.6	0.733	−3.2	1.913	1.797	6980	30.02	+3.5	*E*
104	0.9305	0.784	1.131	70	0.816	+4.1	0.8935	0.8300	7615	24.53	−4.2	0.833	+6.3	0.8752	0.8117	7445	24.00	−6.4	*C*
105	0.4532	0.603	0.964	60	0.586	−2.8	0.4665	0.4058	1942	38.69	+3.3	0.624	+3.5	0.4381	0.3774	1807	36.00	−3.8	*C*
106*a*	0.1733	0.794	1.076	120	0.769	−3.1	0.1791	0.1549	2440	24.92	+3.7	0.764	−3.8	0.1802	0.1560	2461	25.15	+4.6	*C*
106*b*	0.1373	0.698	1.034	115	0.717	+2.7	0.1337	0.1177	1822	23.01	−3.0	0.711	+1.9	0.1349	0.1189	1840	23.22	−2.0	*E*
106*c*	0.0899	0.598	0.931	125	0.587	−1.8	0.0915	0.0655	1021	24.50	+2.4	0.583	−2.5	0.0921	0.0661	1031	24.72	+3.4	*C*
106*d*	0.0790	0.509	0.868	120	0.511	+0.4	0.0787	0.0436	688	26.68	−0.7	0.504	−1.0	0.0798	0.0447	705	27.32	+1.2	*C*
107	1.783	0.797	1.077	95	0.774	−2.9	1.835	1.794	9140	26.69	+3.0	0.765	−4.0	1.856	1.815	9245	27.00	+4.2	*C*
108*a*	0.5065	0.768	1.095	63	0.759	−1.2	0.5125	0.4795	4762	22.32	+1.2	0.788	+2.6	0.4937	0.4607	4577	21.45	−2.8	*C*
108*b*	0.6849	0.826	1.116	63	0.785	−1.5	0.7202	0.6490	6402	30.60	+5.8	0.814	−1.5	0.6947	0.6235	6150	29.40	+1.7	*E*
109	3.682	0.853	1.149	80	0.853	±0.0	3.682	3.602	13187	21.44	±0.0	0.854	+0.1	3.677	3.597	13160	21.40	−0.2	*C*
110	3.726	0.761	1.071	74	0.749	−1.6	3.785	3.692	11810	22.57	+1.7	0.758	−0.4	3.740	3.647	11655	22.28	+0.4	*E*
111	4.203	0.680	1.048	68	0.709	+4.3	4.033	3.783	13685	24.33	−4.3	0.729	+7.2	3.922	3.672	13280	23.62	−7.1	*C*
Compound Condensing Engines—H.P. Cylinders																			
151	1.582	0.786	1.079	100	0.777	−1.1	1.600	1.4988	6820	15.73	+1.2	0.767	−2.4	1.621	1.5198	6915	15.95	+2.6	*C*
152	1.322	0.774	1.060	120	0.749	−3.2	1.366	1.2170	9005	14.09	+3.8	0.744	−3.9	1.375	1.226	9070	14.19	+4.6	*C*
153	0.7188	0.764	1.047	130	0.728	−4.7	0.7542	0.7324	2109	14.97	+5.1	0.728	−4.7	0.7542	0.7324	2109	14.97	+5.1	*C* & *E*
Triple Expansion Condensing Engines—I.P. Cylinders																			
201	2.818	0.508	0.914	38.5	0.470	−7.5	3.048	3.017	5003	15.25	+8.2	0.562	+10.6	2.550	2.519	4180	12.74	−9.6	*C*
High-Speed Single Valve Non-Condensing Engines																			
401	0.4144	0.739	1.027	110	0.710	−3.9	0.4310	0.3362	5115	27.70	+6.1	0.703	−4.9	0.4352	0.3404	5178	28.02	+7.4	*C*
402	0.1792	0.668	0.970	85	0.637	−4.6	0.1879	0.1266	1870	35.10	+7.5	0.631	−5.5	0.1898	0.1285	1897	35.61	+9.0	*C*
403	0.1947	0.675	0.985	74	0.641	−5.0	0.2050	0.1456	2170	35.19	+7.5	0.650	−3.7	0.2021	0.1427	2128	34.48	+5.4	*E*
Simple Locomotive Engines																			
451	1.4267	0.711	1.003	100	0.683	−3.9	1.488	1.2933	7540	29.83	+5.0	0.673	−5.3	1.5110	1.3163	7675	30.38	+6.9	*C*

38 Further investigation shows that there is a relation between the initial quality X_1 and the value of k for adiabatic expansion in the equation for the polytropic curve $PV^k = C$, which, like the experimentally determined relation, is also a linear one. The adiabatic relations of X and k are plotted in Fig. 4 for the pressures used in the tests.

THE METHOD OF APPROXIMATING THE ACTUAL STEAM CONSUMPTION FROM INDICATOR DIAGRAMS

39 When indicator diagrams were first obtained from steam engines, loss from initial condensation was not suspected and it was believed that the steam consumption could be determined from the diagram at the point of cut-off. After the classic tests of Clark and Isherwood, however, the existence and amount of this initial condensation was revealed; but the great difference in the proportion that the initial condensation bears to the total weight of mixture present, either at cut-off or during the expansion, as found in different types and sizes of engines, has prevented any reliable determinations of the actual steam consumption by this method. The steam consumption computed from the diagram, when using saturated steam, is generally from 15 per cent to 50 per cent below the actual consumption.

40 The devising of an accurate method of measuring the actual weight of steam consumed from the diagram has therefore been regarded as impossible. Thurston [2] states that "the steam or water consumption of an engine cannot be exactly ascertained by the use of the indicator" for the reasons mentioned. Most of the other writers on the subject have expressed similar views.

41 Methods of computing the weight or proportion of the initial condensation from the dimensions, type, and speed of the engine considered, as shown by the results of large numbers of tests, have been devised by many.[3] The results obtained from these methods have not been uniform, and do not agree closely enough with the test results to be used with confidence. Moreover, none of these methods is applicable when superheated steam is used. R. C. H.

[1] Principles of Thermodynamics, G. A. Goodenough, p. 191.

[2] Engine and Boiler Trials, R. H. Thurston, p. 237.

[3] A Manual of the Steam Engine, R. H. Thurston, p. 517.
Engineer (London), Escher, 1882.
Relative Proportions of the Steam Engine, L. S. Marks, p. 206.
Proc. Inst. Mech. Engrs., October 1889.
Engineering, Bodmer, March 4, 1892, p. 299.
The Steam Engine, Cotterill, p. 339.

Heck [1] states that the steam consumption computed by the use of his formula should ordinarily show not more than 10 per cent difference from the test results.

42 *The Phenomena Occurring in the Cylinder.* The greatest source of loss in the cylinder is due primarily from practical considerations to the use of a metallic structure. The skin surface of this metal, which is a fairly good conductor of heat, must be heated once every cycle from the temperature acquired from contact with the exhaust steam, nearly up to the temperature of the admission steam, this heating being accomplished by the condensation of some of the incoming steam.

43 The amount of this condensation, measured as the proportion of the mixture present, varies with the size, valve design, relative roughness of the interior surface, temperature range, length of cut-off, speed, location of ports and port passages, quality of the steam supplied, and the jacketing and lagging. It can easily be seen, from the number and relative magnitude of these variables, that the computation of the weight of condensation, by means of a formula which will take these variables into account, can never be an accurate operation.

44 After many examinations into the cases of different types and sizes of engines, with non-jacketed cylinders in good order, and with pressure limits similar to those used with the tests, it has been found that while the initial condensation is subject to the action of ten or more variables, yet the value of n resulting from a given value of X_0 is almost always substantially the same. A few of the applications showing this point will be found in Table 4. Here the cylinder sizes vary from $10\frac{1}{2}$ in. by 12 in. to 34.2 in. by 60 in., the speeds from 27.66 to 263 r.p.m., and the types include slow-speed Corliss, high-speed, and locomotive engines. The possibility of calculating accurately the weight of condensation in these different cases may be easily imagined.

45 The phenomena caused by the presence of the cylinder walls in the class of engines discussed have been found to be divided into two natural classes: those occurring before cut-off, and those occurring after cut-off.

46 The phenomena occurring before cut-off are controlled by the action of the ten or more variables already mentioned, and therefore are subject to all the variation that may occur in any individual case to be examined. For this reason any method of computing the

[1] The Steam Engine, R. C. H. Heck, p. 119.

condensation accurately from the physical facts surrounding the case is open to objection. This method also cannot allow for the use of superheated steam, an increasingly important condition.

47 The phenomena occurring after cut-off are practically independent of all variables except X_e, and the initial pressure and speed. Of these variables, only the value of X_e and the initial pressure have proved to be of material importance in the applications made thus far.

48 This may be summed up by stating that the value of X_e, in any particular case, is subject to the action of many important variables, but that the relation of X_e and n is practically independent of these variables within the limits examined in this investigation.

49 *The Phenomena of Condensation and Re-Evaporation during Expansion.* When adiabatic expansion of initially dry saturated steam takes place, a part of the steam is condensed as the pressure is lowered, the condensed steam giving up its latent heat which is converted into work. When superheated steam is expanded adiabatically, the steam loses its superheat until saturation is reached, after which condensation takes place as in the case of initially dry steam.

50 When, however, the steam is initially composed of a large proportion of water, both being at the same temperature, adiabatic expansion may take place without additional condensation and may even be accompanied by re-evaporation. This fact is due to the large amount of heat contained in the water, a part of which flashes into steam as the pressure is lowered, thus supplying and neutralizing the loss of steam volume by condensation which takes place with steam initially dry. Adiabatic expansion is accompanied by condensation when the initial quality is above the value 0.50 at an initial pressure of 240 lb. per sq. in. absolute, but below the value of 0.50 it is accompanied by re-evaporation. An examination of the temperature entropy diagram for steam will show the values of the initial quality which form the line of demarcation of condensation and re-evaporation during the adiabatic change of state.

51 In the actual engine using saturated steam, as has already been pointed out, some of the incoming steam is condensed in warming up the skin surface of the cylinder walls to approximately the temperature of the incoming steam. When the admission of steam is cut off and expansion commences, the condensation, due to the presence of the cylinder walls, continues in general until, at some point during expansion, the water on the cylinder walls begins to re-

evaporate at such a rate that the weight of steam present at the end of expansion is greater than that which was present at cut-off.

52 To show the effects and extreme values of condensation and re-evaporation during expansion, there have been prepared in Table 5 two examples, using the average results of the tests run.

TABLE 5 CONDENSATION AND RE-EVAPORATION OF STEAM DURING EXPANSION

Example number	1	2
Initial quality, parts of unity	0.540	0.950
Pressures, Lb. per Sq. In. Absolute		
Initial	145.0	145.0
Final	20.0	20.0
Value of n in Equation $PV^n=C$		
Adiabatic expansion, from Table 20	1.082	1.133
Actual expansion in engine tested, from Fig. 6	0.900	1.230
Volume of Steam Present, Cu. Ft. (Volume of water neglected)		
Initial	1.00	1.00
Final, adiabatic expansion	6.24	5.75
Final, curve of constant steam weight	6.44	6.44
Final, actual expansion in engine tested	9.04	5.00
Weight of Steam Present, Lb.		
Initial, plus water	0.594	0.338
Initial, steam only	0.321	0.321
Final, adiabatic expansion	0.311	0.287
Final, curve of constant steam weight	0.321	0.321
Final, actual expansion in engine tested	0.450	0.249
Quality of Steam at Final Pressure, Parts of Unity		
Adiabatic expansion	0.524	0.849
Curve of constant steam weight	0.540	0.950
Actual expansion in engine tested	0.757	0.736
Condensation or Re-evaporation, Parts of Unity		
Apparent re-evaporation	0.217	
Real re-evaporation	0.233	
Apparent condensation		0.214
Real condensation		0.113

53 Example 1 is a condition which obtains in the engine tested when using saturated steam at about 140 lb. gage pressure with a length of cut-off of about 3 per cent. All values of the qualities mentioned are portions of the total weight of mixture in parts of unity. The value of X_c is 0.540, a low value, yet one which often obtains in small engines. If this steam were expanded adiabatically

to the back pressure, 20 lb. absolute, the resulting quality would be 0.524, giving a condensation of 0.016: The expansion which actually takes place in the engine tested under these conditions results in a final quality of 0.757, showing that the apparent re-evaporation from the value of X_e has been 0.217. However, the steam mixture in expanding *did* expand adiabatically in order to give up heat to work, but the actual or what might be called the gross expansion, was changed in character by the re-evaporation of a large proportion of the water present, due to the return of heat from the cylinder walls and the consequent flashing into steam of part of the water when the pressure and the temperature were lowered. The real re-evaporation, measured by its effect upon adiabatic expansion, has been the difference between 0.757 and 0.524 or 0.233.

54 The actual expansion in this case has been the result of two factors which worked simultaneously: adiabatic expansion and the return of heat from the cylinder walls to the mixture. The first factor, adiabatic expansion, as already explained, is itself the result of two neutralizing or opposing conditions, i. e., the condensation of initially dry steam during expansion, and the relatively smaller amount of re-evaporation of water initially in the mixture due to the liberation of its excess of heat when the pressure and temperature were lowered. The net result of the two conditions of this adiabatic expansion, however, was a condensation. The second factor is the large amount of re-evaporation due to the return of heat from the surface of the cylinder walls to the condensed steam, amounting in Example 1 to 0.233, or, roughly, there has been re-evaporated during expansion $\frac{1}{4}$ of the entire weight of mixture present.

55 Example 2 shows conditions which obtain in the engine tested when served with steam superheated about 125 deg. fahr. at about 140 lb. gage pressure, with a length of cut-off of about 45 per cent. The value of X_e is 0.950, a very high value for this class of engine. The quality after adiabatic expansion would be 0.849, a condensation of 0.101. Where values of X_e are as high as 0.950, however, no re-evaporation takes place in practice, but condensation continues throughout expansion. After expansion in the engine tested, the quality would be 0.736, showing much greater condensation than that due to adiabatic expansion alone. The apparent condensation has been 0.214, but the real condensation, measured by its effect upon adiabatic expansion, has been 0.113.

56 The actual expansion in Example 2, as in Example 1, has been

the result of two factors: adiabatic expansion and the further abstraction of heat during the whole expansion by the cylinder walls. Heat is abstracted during expansion by the cylinder in the engine tested at all values of X_c above 0.85, thus giving values of n higher than the adiabatic value k.

57 The phenomena of condensation and re-evaporation during expansion are the causes of the relations existing between X_c and n in the cylinders of steam engines. The two examples given show values obtained in extreme cases which illustrate very well the effect of X_c upon the character of the expansion, and therefore upon the value of n, and show the range of values that n assumes in one engine due to a change in the value of X_c.

58 *The Relation of X_c and n.* The relation of X_c and n, for the same engine, has been shown by the author's tests to be very definite under the same conditions of pressure and speed. This relation, or dependence of n upon X_c, however, is not seriously affected by changes of pressure and speed within the limits of the tests.

59 The ordinary speeds of similar types of engines, 70 to 120 r.p.m., do not affect the relation seriously enough to be taken into account when examining such cases, because of the twofold character of speed in its influence upon the action of the cylinder walls.

60 The engine experimented upon was operated at 120 r.p.m., and had a stroke of 2 ft. Other engines of this class run at speeds as low as 70 r.p.m., but have strokes of 5 or 6 ft. Cylinder condensation is not dependent upon rotative speed alone, but is also influenced by the piston speed, as determined by the length of the stroke. On account of different lengths of stroke, different engines cannot be compared on the basis of rotative speed. Thus while the small engine tested has a rotative speed of 120 r.p.m., its piston speed is only 480 ft. per min. In large engines, while the rotative speed may be only 70 r.p.m., yet, with strokes of 6 ft., the piston speed is 840 ft. per min. What the small engine gained by higher rotative speed, the large engine made up in a measure by higher piston speed.

61 After taking into account the two parts of which speed is composed, it is found that the speeds of stationary engines of the type tested are in substantially the same range. On account of this fact, only the results of the tests run at 120 r.p.m. have been used in the applications made at present.

62 The relation of X_c and n has been found to be practically

independent of cylinder size. This statement is true for non-jacketed cylinders and for pressures in the range examined.

63 It appears, therefore, that this method of approximating the value of X_e at cut-off from experimentally determined relations, and thus accounting for the initial condensation, is upon much surer ground than any method of computing condensation from the physical facts surrounding the case. It approaches the problem from the side where the phenomena occurring are practically independent of all the variables mentioned. This fact adds greatly to the accuracy obtained, and with the applications given in Table 4 shows that an accurate and reliable method of approximating the actual steam consumed from the indicator diagram alone has been devised.

64 This method is free from several objections to which tests are open. It measures the consumption in one revolution, and is, therefore, practically measuring a *rate* instead of a *quantity.* The only data needed for an approximation are one set of indicator diagrams, taken simultaneously, the constants of size and clearance, and the speed of the engine tested. No interruption of any kind in the routine of a plant is caused, and the expense incurred is not to be compared with that of an equally accurate test. The method is accurate enough for almost all purposes except guarantee tests subject to bonus and forfeit contracts. In the case of locomotives on the road, it is the *only* possible method of approximating the steam consumption of the main engines, due to the use of steam by the air-pump, train-heating system, blower, generator sets, safety valves, whistle, blow-off valves, and leaks. The same is true of marine engines, where many auxiliaries are supplied with steam from the same boilers, and exhaust into the same surface condensers. The method is especially useful for non-condensing engines, where the boiler-feed measurement method is the only practicable one. Steam consumption may be obtained as often as is desired instead of probably once in an engine's life.

65 *Limitations.* The relations of X_e and n given in Fig. 6 are applicable, however, only to non-jacketed cylinders exhausting at very close to atmospheric pressure. When the back pressure is raised to 30 lb. absolute, for instance, there is a new series of relations existing for the same initial pressure, due to a different temperature range in the cylinder and the consequent alteration of the phenomena occurring after cut-off. Steam jackets also alter the phenomena occurring after cut-off, and therefore have to be examined separately for the relations of X_e and n.

66 Since this method rests entirely on the indicator diagram, great care must be observed in taking these diagrams. The indicator itself must be an accurate instrument in the best possible condition. The indicator connections must be short and direct. An extensive investigation by W. F. M. Goss [1] shows that the long and indirect pipe connections materially alter the form and character of the expansion curves. A correct reducing motion, free from lost motion, must be used so as to reproduce the actual expansion. The arrangement of having one indicator at each end of the cylinder is always to be preferred.

67 The applications of this method must be made with judgment and care. If large leakage exists only an approximate solution can be obtained, as certain assumptions, treated in the section on leakage, have to be made. The various steps involved in the use of the method must be thoroughly comprehended to give satisfaction.

68 *Application of the Method.* The relations of X_e and n, as determined for various pressures at constant speed from the engine tested in the laboratory of the University of Illinois were plotted in the form of the chart shown in Fig. 6.

69 The next step was to examine, with certain restrictions, the tests of other engines, and to compare the relations of X_e and n with those given in Fig. 6. The restricting conditions imposed were: (*a*) that the tests should come from reliable sources; (*b*) that the data supplied should be complete enough for computing the quantities needed for comparison; (*c*) that the cylinders should be non-jacketed; (*d*) that the diagrams furnished should be representative of average conditions; (*e*) that the back pressure in the cylinder examined should be practically atmospheric; (*f*) that no large leaks should exist.

70 The values of X_e, n and p were first found from the set of diagrams to be examined. Next, the values of n and p were located in Fig. 6 and the corresponding value of X_e found, as obtained in the tests. The value of X_e obtained from the chart and that obtained from the test examined were compared, and the steam consumption, as computed by the value of X_e taken from the chart, was obtained.

71 The results of tests which fulfilled the conditions imposed are given in Table 4. Four distinct classes of engines were examined. These include simple Corliss, two-valve and four-valve types, the high-pressure cylinders of compound engines, the intermediate pressure cylinders of triple expansion engines, high-speed and simple

[1] Trans. Am. Soc. M. E., vol. 17, p. 398.

locomotive engines. The sizes range from $10\frac{1}{2}$ in. by 12 in. to 34.2 in. by 60 in., and the speeds from 263 to 27.66 r.p.m.

72 The final results, given in columns 24, 29, 31 and 36 of Table 4 were averaged (analysis 201 excepted) and the averages are given as follows:

Average Difference from Test Results	Value of X_c, Per Cent	Actual Steam Consumption, Per Cent
Approximation from Chart—Fig. 6		
Irrespective of sign	3.06	3.72
Higher (+) or lower (−) than test results	−1.93	+2.50
Approximation from Equation $X_c = 1.245\,n - 0.576$		
Irrespective of sign	3.32	3.98
Higher (+) or lower (−) than test results	−1.32	+1.91

Analysis 201 shows an application, the conditions of speed and cut-off pressure of which are far outside of the limits examined. The values given were obtained by extrapolating as straight lines the lower portions of the curves of constant value of X_c in the chart of Fig. 6. Although the speed is only 27.66 r.p.m. and the cut-off pressure only 38.5 lb. absolute, the value of X_c by chart was determined as 0.470, while the value by test is 0.508, a difference of 7.5 per cent, based on the test value of X_c. This application is given to show that an extrapolation of the method to unusual conditions of speed and cut-off pressure does not lead to absurd results, although it is not nearly as accurate as the applications to speeds higher than 50 r.p.m.

73 The results of the applications made up to the present, with the restricting conditions imposed, tend to show that the steam consumption of engines may be approximated from the indicator diagram to within an average difference of less than 4 per cent from the test results. Individual examples, however, may show as much as 8 per cent difference in rare cases.

CONCLUSIONS REGARDING STEAM CONSUMPTION FROM DIAGRAMS

74 The following conclusions have been drawn from the results of the investigation, as applicable to non-jacketed steam cylinders in good physical condition exhausting at or near atmospheric pressure, and with the limitations imposed as already given:

a At a given initial pressure and speed of engine, there is a definite relation existing between X_e and n, in any one cylinder, which is practically independent of the cut-off position.

b This relation is practically independent of cylinder size and of engine speed; it is therefore applicable to other cylinders of the same type.

c By means of the experimentally determined relations of X_e and n, the value of X_c may be approximated from the average value of n obtained from the expansion curves of one set of indicator diagrams, taken simultaneously; therefore the actual weight of steam present in one revolution may be approximated.

d The actual steam consumption may be obtained by this method from the indicator diagram to well within an average of 4 per cent of the amount consumed as measured by test.

e This method has the following advantages not possessed by tests: it is more accurate than the average test, and is the only accurate method available for testing certain classes of engines; it virtually measures an instantaneous rate instead of an average quantity over a long time, and thus enables a large number of points to be obtained for a water-rate curve; it permits of making tests at frequent intervals instead of once in the engine's life; the expense is not to be compared with that of an equally accurate test; it involves no change in the routine of the plant tested.

THE LOGARITHMIC DIAGRAM APPLIED TO ALL ELASTIC MEDIA

75 *The Form of Expansion and Compression Curves from Practice.* About 300 indicator diagrams from 50 engines using steam, gas, air, and ammonia have been examined [1] to investigate the form and character of the expansion and compression curves. It may be stated as a result that the polytropic law $PV^n = C$ holds for all elastic media with certain exceptions which have been studied and the causes treated.

76 An example of two logarithmic diagrams showing how perfectly the law holds is given in Fig. 10. These diagrams were constructed from two indicator diagrams shown in Fig. 9 taken from the Corliss engine tested.

[1] See Appendix for discussion.

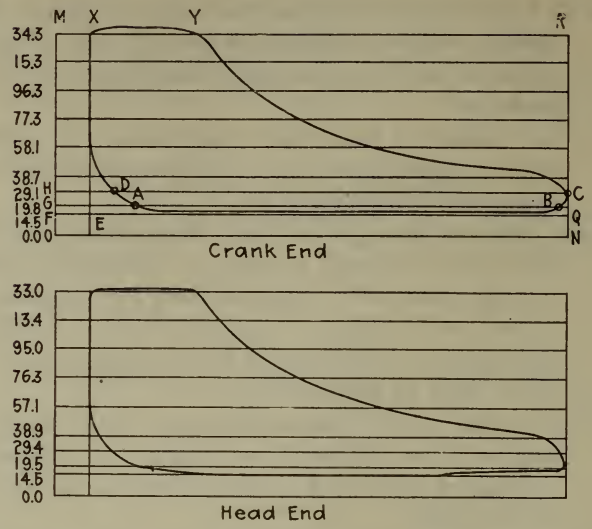


FIG. 9 DIAGRAMS TAKEN FROM TEST 30, TABLE 1, 12 IN. BY 24 IN. CORLISS ENGINE

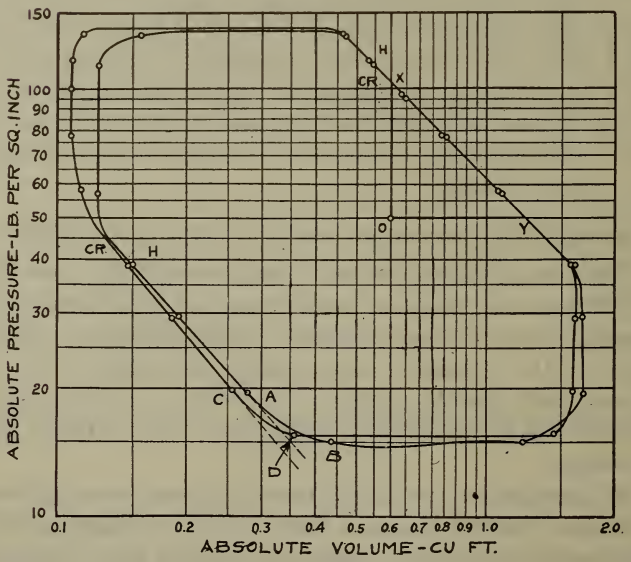


FIG. 10 LOGARITHMIC DIAGRAMS PLOTTED FROM FIG. 9

77 *Mathematical Relations of the Law,* $PV^n=C$. The equation of the polytropic curve, $PV^n=C$, when plotted on rectangular cross-section paper, gives a curve depending for its form and position on the values of P, V, n, and C. When this curve is plotted on logarithmic paper it becomes a straight line depending for its slope on the value of n and for its position upon the value of C. The relations for such curves are given:

$$PV^n=C$$

Taking the logarithm of both terms

$$\log P+n \log V=\log C$$

Transposing

$$\log P=-n \log V+\log C$$

This equation is of the form of the straight line

$$y=mx+b$$

where

$$y=\log P$$
$$m=-n$$
$$x=\log V$$
$$b=\log C.$$

Thus $m=-n$, the slope, or measure of inclination of the line to the axis $\log V$. In Fig. 10, for example, at a point X on the line

$$\log P=-n \log V+\log C$$

draw OX parallel to the axis $\log P$, and draw OY parallel to the axis $\log V$. The slope of the line will be the value of the ratio $\frac{OX}{-\overline{OY}}=-n$.

OY is negative, being measured to the left, giving n its negative sign.

78 *Use of the Logarithmic Diagram.* The logarithmic diagram forms the basis of the methods of analyzing the cylinder performance of reciprocating engines which are developed in detail in the following pages.

79 These methods apply only, however, to the logarithmic diagrams derived from the cylinders of reciprocating engines using an elastic fluid for the working medium and having, as a part of the cycle of operation, an expansion, a compression, or both.

80 The figures of one set of indicator diagrams and the corresponding set of logarithmic diagrams are numbered the same, but the letters *a* and *b* are used in addition to the figure number to denote the indicator and logarithmic diagrams, respectively.

RATIONAL METHOD OF APPROXIMATING CLEARANCE

81 In the cases of the great majority of the PV-diagrams which were examined,[1] the expansion and compression curves became straight lines in the logarithmic diagram, showing that the law $PV^n = C$ was applicable, or in other words, that n was a constant for one curve. The clearances furnished with the diagrams examined had been carefully found by the displacement method.

82 It was desired to see what forms the lines assumed when the clearance was taken larger or smaller than the measured quantity. The diagram shown in Fig. 11-a, taken from a 42 in. by 60 in. gas engine, was used for this purpose. The true clearance, measured as 18.0 per cent, was used in the full logarithmic diagram of Fig. 11-b. Trials were made with clearances assumed as 14.0, 16.0, 20.0, and 22.0 per cent of the piston displacement. With the true clearance of 18.0 per cent, the curves became almost perfectly straight lines, while with the values of clearance less than 18.0 per cent, it is seen that the lines become bent to the left, and with values of over 18.0 per cent the lines become bent to the right. Hence the straight line for the value of 18.0 per cent is the transition between the family of curves bending to the left, representing a clearance smaller than the real value, and the family of curves bending to the right, representing a clearance larger than the real value.

83 The practical significance of this fact is that there is now available a rational method of approximating the clearance of any cylinder using an elastic medium, which has, as a part of the cycle of operation, an expansion or a compression. This method is based on the fact, already mentioned, that in practice all elastic media, except under certain exceptional conditions, obey substantially the law $PV^n = C$, when subject to change of state, and therefore become straight lines in the logarithmic diagram.

84 *Graphical Method of Approximating Clearance.* The graphical method of approximating clearance requires only the scale of the indicator spring to be known, and the atmospheric line to be drawn, in order to locate the zero line of pressure. The exact order of procedure necessary to make a trial, and the degree of accuracy obtained in any given case, is shown in detail in Par. 93 for a $25\frac{5}{8}$ in. by $37\frac{5}{8}$ in. gas engine. All that is necessary is to assume different values of clearance, and plot the logarithmic diagram for each assumed value. The straight line position of the curves is found by

[1] See Appendix for discussion.

trial and error to lie between the two diverging families of curves representing too small and too large clearance.

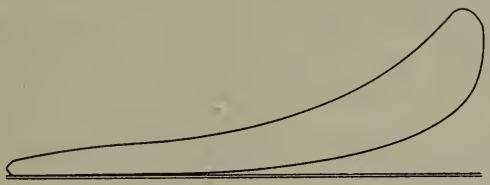


FIG. 11-a DIAGRAM FROM LARGE GAS ENGINE

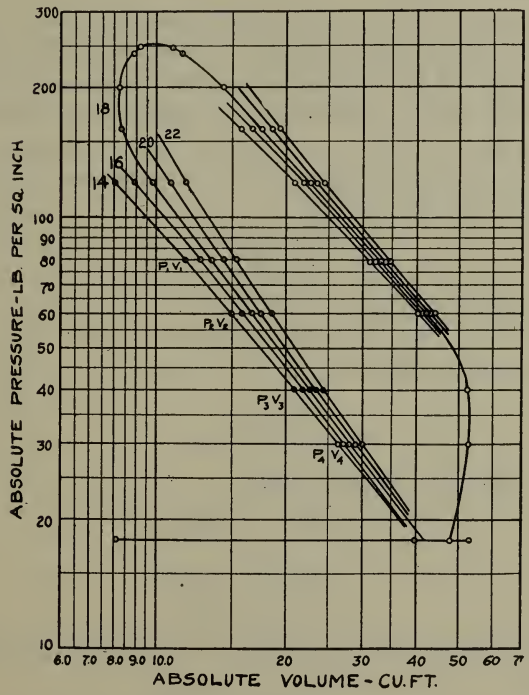


FIG. 11-b TOD 42 IN. BY 60 IN. GAS ENGINE, BLAST FURNACE GAS

85 It follows also from the curves shown by Fig. 11-b that the clearance being known, the scale of the spring used may be obtained in the same manner if the atmospheric line is given.

86 *Mathematical Method of Approximating Clearance.* The results obtained from the graphical method of trial and error may also be accomplished by the use of the purely mathematical process upon which the method depends.

87 The law $PV^n=C$ holds, thus n is a constant for any part of the curve. When the wrong clearance is used, the law $PV^n=C$ does not hold, and n varies from point to point. In the graphical method, trials of various values of clearance are made, until the curve becomes approximately a straight line; this resultant straight line is, therefore, the law $PV^n=C$, in which n is a constant for all parts of the curve. The one condition necessary to be fulfilled, therefore, is that n be constant for all parts of the curve, but not of any particular value.

88 To illustrate the use of the mathematical method in Fig. 11-b, let us assume several points, as P_1V_1, P_2V_2, P_3V_3, and P_4V_4 at various intervals on one of the curves, as on the compression curve at the clearance value 14.0 per cent. It is desirable for convenience to locate the points at about equal intervals, as shown. The law $PV^n=C$ is assumed to hold. Then, for two points, P_1V_1 and P_2V_2, called group a, we have

$$P_1V_1^n=C$$
$$P_2V_2^n=C$$

Equating these, we obtain

$$P_2V_2^n=P_1V_1^n$$

Transposing and dividing

$$\left(\frac{V_2}{V_1}\right)^n=\frac{P_1}{P_2}$$

Taking the logarithm of both sides

$$n\ (\log V_2-\log V_1)=\log P_1-\log P_2$$

Whence

$$n_a=\frac{\log P_1-\log P_2}{\log V_2-\log V_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

In the same manner for the points P_3V_3 and P_4V_4, called group b, is obtained

$$n_b=\frac{\log P_3-\log P_4}{\log V_4-\log V_3} \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

For the correct value of clearance, the following condition must be fulfilled by trial and error

$$n_a=n_b \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

89 The values of the logarithms of the coördinates of all the

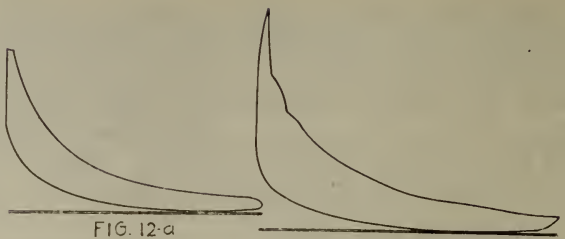
points are then found, and the values of n_a and n_b computed. If the points are located in the order shown, then with too small a clearance, n_a is lower in value than n_b. A larger value of clearance is then assumed, the operation being merely to add a constant number to the values of V_1, V_2, V_3, and V_4. The process is repeated until the value of n_a becomes practically equal to n_b. When the clearance is assumed too large, n_a becomes higher in value than n_b, indicating that the true value has been passed.

90 The trial by the mathematical method is neither as accurate nor as short as the graphical method. It is not as accurate because the points assumed may not be representative. When this is the case, the graphical method allows judgment to be exercised in selecting the straight-line position, thereby eliminating irregularity of points.

91 The question arises as to whether the form of the lines due to wrong clearance can be distinguished from the form due to leakage or to "hooks," on the logarithmic diagram. This case is treated in Par. 126.

92 The curve of the PV-diagram nearest the clearance space, or the compression curve in Fig. 11-a, is generally the better guide in the graphical trials. This is well shown in Fig. 11-b. A given difference in the values of clearance used for trial causes more horizontal variation in the position of the compression curve than in the expansion curve. This fact allows closer locations of the straight-line transition region to be made from the compression curve than from the expansion curve.

93 *Examples.* It was desired to determine the clearance of the diagram shown in Fig. 12-a. From general knowledge of this class of engines a trial by the graphical method was made in Fig. 12-b with the clearance assumed as 12.4 per cent, a value purposely assumed as being too small. This value is seen, by the bending of both curves to the left, to be much too small. Trials were, therefore, made with the clearance assumed as 13.8, 15.1, and 16.0 per cent of the piston displacement. The values of 15.1 per cent gave practically straight lines for both the expansion and compression curves, while the value of 16.0 per cent shows that the lines have begun to bend to the right, indicating too large a clearance. By inspection, it will be seen that the region of fairly straight lines may be located as lying between the values of about 14.5 per cent to 15.5 per cent. The clearance is, therefore, selected as 15.0 per cent, a value which



DIAGRAMS FROM GAS ENGINES FOR PRODUCER AND ILLUMINATING GAS RESPECTIVELY

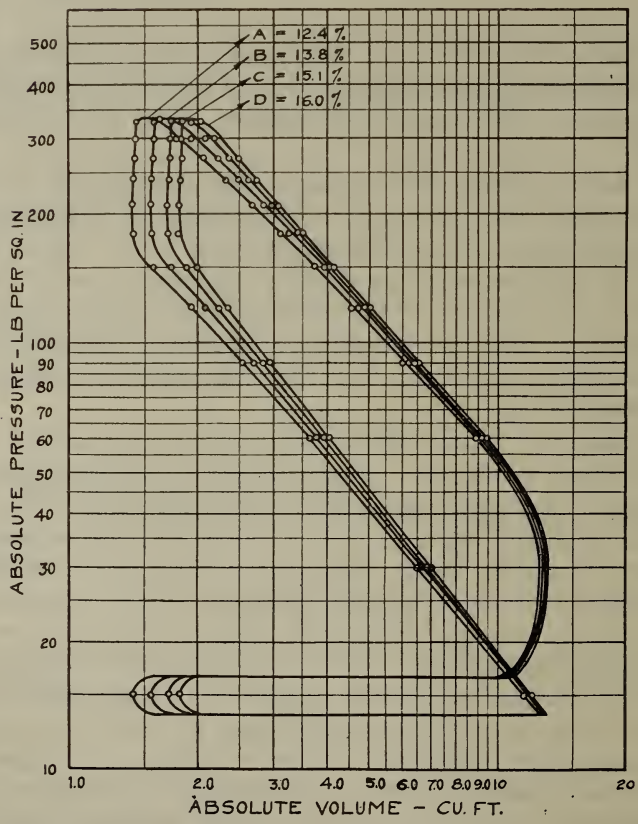


FIG. 12-b KOERTING 4-CYCLE 25⅜ IN. BY 37⅝ IN. PRODUCER GAS ENGINE

may be high or low by not more than 4 per cent in this case. This clearance value, 15.0 per cent, is a common value for engines of this class.

94 The graphical method is more accurate for large clearances, measured in per cent of the piston displacement, than for small ones.

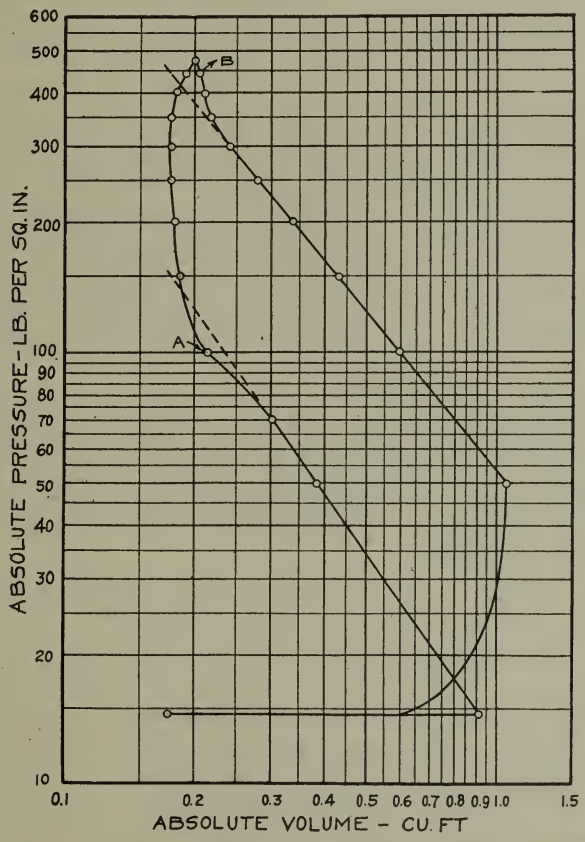


Fig. 13-b Gas Engine 10 in. by 19 in. for Illuminating Gas

The closeness of location of the straight-line region, lying between the two families of diverging curves, will be found to be within 5 per cent to 10 per cent of the clearance volume, for values of clearance between 20 per cent and 2 per cent, respectively, of the piston displacement.

RATIONAL METHOD OF LOCATING THE STROKE POSITION OF CYCLIC EVENTS

95 It is often desirable to know at what part of the stroke the cyclic events occur. This knowledge can be best obtained from the PV-diagram. For ordinary purposes, these events can be closely located by inspection on the PV-diagrams themselves, in most cases; thus, on a diagram from a Corliss engine, cut-offs may generally be located to within $\frac{1}{16}$ in., measured along the length of the diagram.

96 The actual beginning of true compression, however, can *never* be accurately located on the PV-diagram. True compression, unaffected by leakage, begins after the exhaust valve, in closing, has acquired enough seal to prevent leakage. The point of the beginning of true compression is generally at least 5 lb. above the back pressure. The point at which leakage ceases cannot be located on the PV-diagram because the curve of true compression, and the curve during the time the valve has insufficient seal, are of the same direction of curvature, and are not reverse curves as in the general case of admission and expansion.

97 The fact that expansion and compression of a constant weight of medium takes place[1] according to the law, $PV^n = C$, thus becoming straight lines in the logarithmic diagram, enables us to locate cyclic events very closely, even in cases where they cannot be detected at all in the PV-diagram.

98 An example is shown in Fig. 20-a, containing locomotive PV-diagrams taken at short cut-off and high speed. The events of cut-off, release, compression, and lead are very difficult to locate on such diagrams. These events are located on the logarithmic diagram in Fig. 20-b by noting when the expansion and compression curves become straight, indicating a constant weight of steam mixture.

99 A sufficient number of points are plotted to show clearly the direction of the diagram near the events desired. Thus these events, even though obscure in the PV-diagram, may be located to well within about $\frac{1}{16}$ in. in the logarithmic diagram, this length being equivalent to about $\frac{1}{32}$ in. when retransferred to the PV-diagram itself.

100 The use of this method has one great advantage in that it largely eliminates the variable element of personal judgment. It is a common occurrence to see PV-diagrams where two persons have

[1]See Appendix for discussion.

located an event such as cut-off, $\frac{1}{8}$ in. apart, each location being the best judgment of the person doing the work. The logarithmic diagram will at all times give closer locations of events for these reasons than will the PV-diagrams.

101 The method also allows the point of true compression to be located, the location of which is practically impossible in the PV-diagram.

RATIONAL METHOD OF DETECTING LEAKAGE

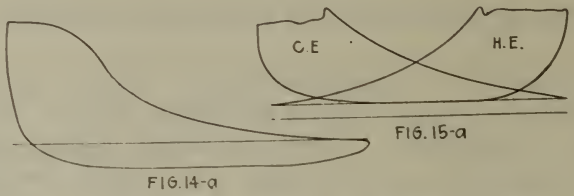
102 The law $PV^n = C$ is applicable only to cases where the weight of the working medium remains practically constant during any expansion or compression. When this weight changes materially, either by leakage into, or out from, the cylinder containing the medium, the resulting expansion or compression no longer obeys the law, and it becomes a curve on logarithmic cross-section paper. This fact is very clearly shown in the curves of the logarithmic diagram derived from cylinders in which large leaks were known to exist.

103 *Examples of Known Leakage.* The first case, shown in Fig. 13-b, occurred in a 10 in. by 19 in. gas engine, intended for producer gas, but using illuminating gas at high compression. The piston, a single-acting trunk type, allowed a large leak, clearly detected by the noise of escaping gas, at the beginning of the combustion stroke. Both the compression and expansion curves show the effect of this leak in a clear manner when transferred to the logarithmic form. After that portion of the stroke was reached where no sound of leakage was heard, the two curves became straight lines. This indicated very clearly that the effect of leakage, if appreciable, may be detected in the form of the curves of the logarithmic diagram.

104 The second case, shown in Fig. 14-b, is from a 14 in. by 35 in. Corliss engine. The knowledge of the leaky condition of the piston and valves came from the engineer in charge.

105 The expansion and compression lines indicate by their form at the upper ends, a large leak from the cylinder, or through the exhaust valve. The lines also show, by the rising of the curves at the lower ends, a considerable addition to the steam in the cylinder during expansion and compression. This steam could come only from a large leak in the steam valve. The seven other diagrams taken from this same engine all showed the effect of leakage in a similar manner.

106 The third case of known leakage, shown in Fig. 15-b, is from an $11\frac{1}{4}$ in. by 22 in. double-acting ammonia compressor. This cylinder was known to be in a very bad condition as regards wear and leakage of piston and valves. The re-expansion curves, by the enormous amount of re-expansion shown, indicate large leakage into the cylinder during this operation. The lower part of the com-



DIAGRAMS FROM CORLISS ENGINE AND AMMONIA COMPRESSOR

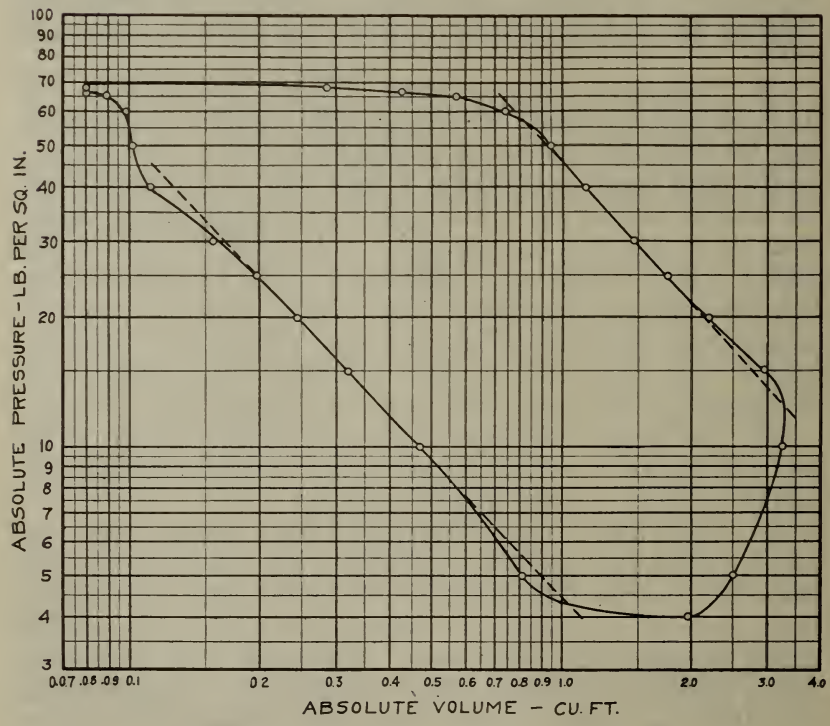


FIG. 14-b CORLISS ENGINE 14 IN. BY 35 IN.

pression curves, by rising, indicates leakage into the cylinder, either past the piston or through the discharge valves. The upper part of

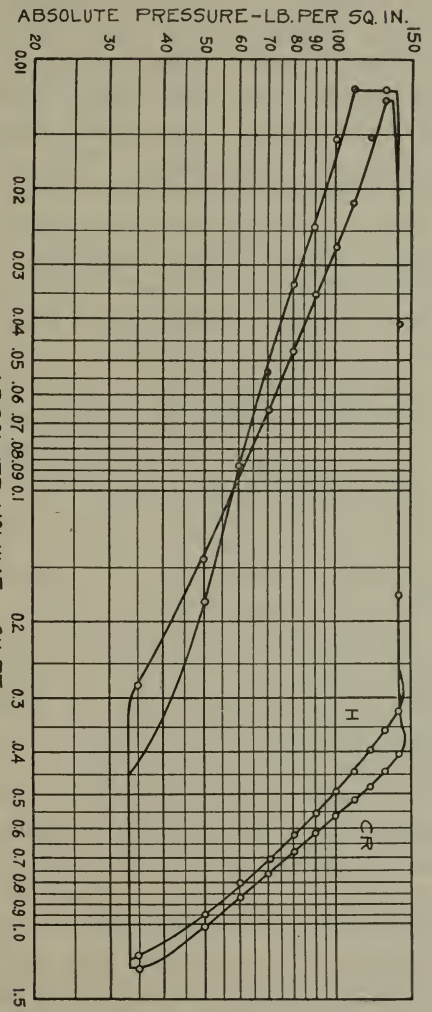


Fig. 15-b Ammonia Compressor 11¼ in. by 22 in.

these curves indicates leakage from the cylinder, either past the piston or through the suction valves.

107 These three examples show abnormal conditions which are comparatively rare. Very smooth curves may be obtained in the

PV-diagram even if there is large leakage taking place. This is seen by referring to Fig. 14-a, both the expansion and compression curves being fairly regular. The logarithmic diagram, however, shows clearly, in connection with the discussion and the examples shown, that large leakage of two kinds was taking place during expansion and compression. Leakage which occurs during admission or during exhaust has no effect upon the lines of the diagram as the weight of the medium is continually changing.

108 *Method of Detecting Leakage.* When leakage occurs in a cylinder, it is seldom found that only one source of leakage exists. Leakage is usually the result of wear, which affects most of the possible sources of leakage in about an equal proportion. As a result, several leaks are generally affecting the curves. This is the case in Figs. 14-b and 15-b, where leakage was taking place both into and out from the cylinders.

109 In discussing leakage, it must be kept in mind that difference in pressure between two regions is the cause of this phenomenon. In the steam engine, there are three pressures which must be considered, i. e., the pressure in the steam chest, in the cylinder at the point discussed, and in the exhaust passage. Leakage, being due to difference of pressure, becomes material only when this difference becomes considerable. Thus leakage into, or out from a steam cylinder has been found to occur in most cases only when the pressure difference is over about 20 lb. In Fig. 14-b, the leakage into the cylinder, shown by the lower parts of the lines, begins to occur at about 25 lb. absolute, or 35 lb. lower than the pressure at admission. The leakage out from the cylinder, shown by the upper parts of the lines, ceases to occur at a pressure of about 40 lb. absolute for the expansion curve, and begins to occur at about 25 lb. absolute for the compression curve. The difference of pressure between the steam in the cylinder and that in the exhaust passage is about 35 lb. in the first case, and about 20 lb. in the second case.

110 This fact, founded on many diagrams analyzed enables us to divide the expansion and compression lines roughly into three equal parts on the logarithmic diagram (when these lines extend from the initial pressure nearly to the back pressure): (*a*) the upper third, influenced by leaks *out from* the cylinder; (*b*) the middle third, practically *uninfluenced* by leakage; (*c*) the lower third, influenced by leakage *into* the cylinder. Thus fairly reliable values of n, free from the effect of leakage, may be obtained from the middle third of the lines.

111 Returning to Fig. 14-b, both the lines indicate leaks out from the cylinder. This can occur either past the piston or through the exhaust valves. The piston generally becomes leaky sooner than Corliss exhaust valves, and, in this particular engine, one of the piston rings was found upon examination to be broken. When diagrams from both ends of the cylinder are available, piston leakage causes nearly an equal effect on the expansion curves of both ends. The leakage into the cylinder can come from only one source which can influence the curves, i. e., the steam chest. The effect of this leak is seen in both lines in the lower thirds.

112 In Fig. 15-b, both forms of leakage are shown in the ammonia compression curves. The lower thirds show leakage into the cylinder, either through the discharge valves or past the piston. The upper thirds of the lines show large leakage from the cylinder, caused by the condition of either the suction valves or the piston. Ordinarily, it is not possible to distinguish between two leaks occurring in the same third of the curves.

113 *Approximation of the Volume Which Leaked.* Fig. 13-b is an example where only one kind of leakage is present. Here, the piston alone leaked badly at the commencement of the stroke. The effect of this leak is seen in the upper third of both lines.

114 When only one kind of leakage exists it is possible to compute with fair accuracy the volume of leakage taking place during expansion or compression. The lines are extended, as shown in Fig. 13-b, giving the lines of constant weights of the medium. The volume of gas that had leaked during compression, up to 100 lb. absolute pressure, is then seen to be 0.014 cu. ft., or 6.3 per cent of the volume remaining. The volume of gas, measured at the pressure of 450 lb. absolute, that leaked after combustion during expansion, is seen to be 0.032 cu. ft., or 18.7 per cent of the volume remaining after the leakage stopped.

115 The leakage that took place during combustion at the end of the stroke cannot be computed, but it can be estimated by making the assumption that this leakage was proportional to the mean rate of leakage shown by the two curves, and that its duration was the time interval occurring between the point *A* and the point *B*.

116 The important result that is attained by this method is not, however, the approximation of leakage, but the knowledge that it is taking place, so that it can be located and stopped.

117 *The Use of the Method in Testing for Maximum Economy.* Many engines are sold and their prices fixed on the basis of their test

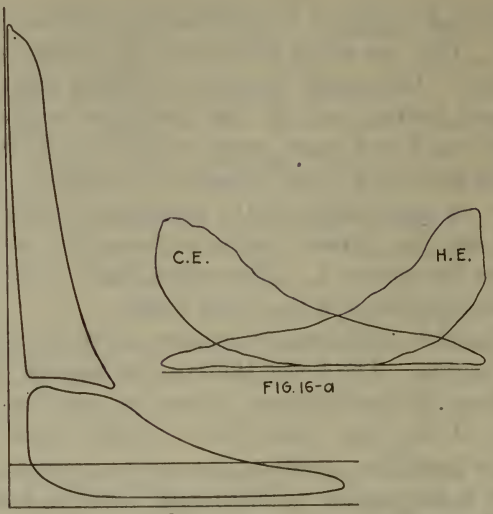


FIG. 16-a

FIG. 17-a

DIAGRAMS FROM HIGH-SPEED ENGINE AND COMPOUND SUPERHEATED STEAM ENGINE

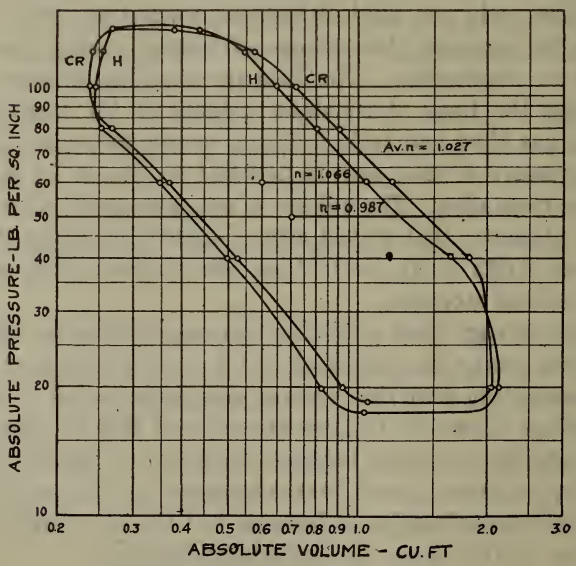


FIG. 16-b IDE 16 IN. BY 16 IN. HIGH-SPEED STEAM ENGINE

performances. The importance to the manufacturer of being able to eliminate leaks during this test does not have to be emphasized. Does the engineer in charge of the test know whether the engine is tight under regular operating conditions?

118 All of our present knowledge of leakage is an inference drawn from the leakage "standing." Nobody knows whether an engine

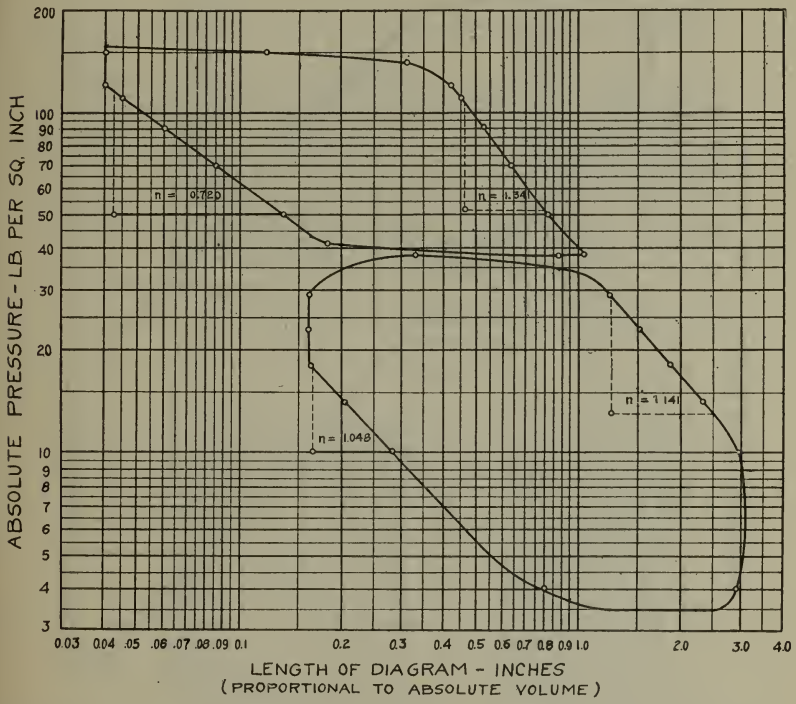


Fig. 17-b Rice and Sargent 16 in. by 28 in. by 42 in. Steam Engine Using Highly Superheated Steam

that is tight "standing" leaks when in operation, or vice versa.

119 This method should be applied to all engines about to undergo any test where maximum economy is the object desired.

120 The results of an analysis made in connection with this investigation by means of logarithmic diagrams of a large number of PV-diagrams, indicate that the majority of engines, in good condition, are practically tight as regards leakage into or out from the cylinder.

121 Most leakage is the result of wear and tear due to long and hard use. After the wear and tear has become marked, it is the custom to rebore the cylinder and to resurface the valves and valve-seats. Some railroad companies overhaul the cylinders and valves

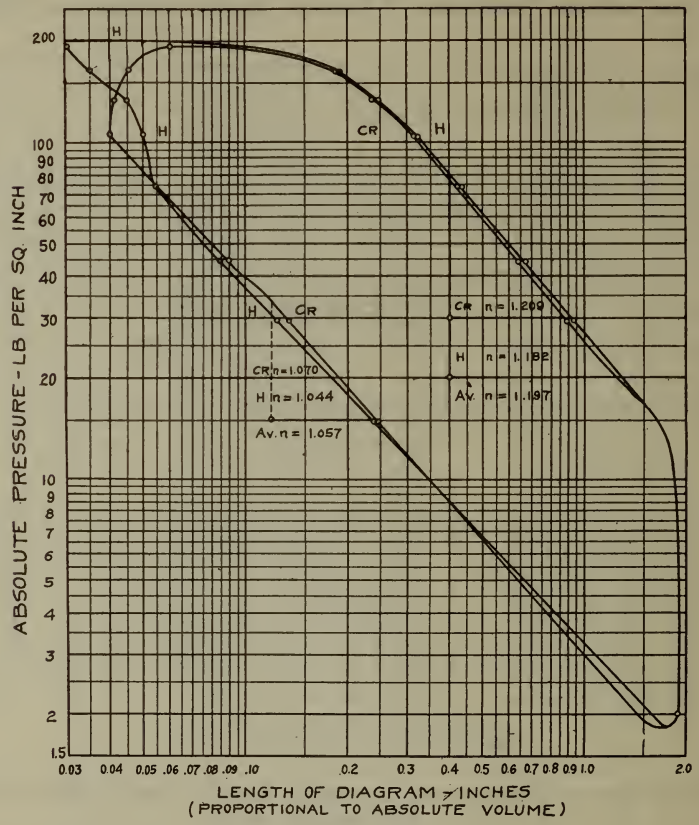


FIG. 18-b STUMPF 23½ IN. BY 31½ IN. UNI-DIRECTIONAL FLOW STEAM ENGINE USING SUPERHEATED STEAM

of locomotive cylinders at regular intervals of, for instance, 150,000 miles of travel. Some cylinders in stationary plants are rebored at equal time intervals of some four or five years each.

122 As a pure question of economy, other things not considered, general repairs of cylinder and valves should take place when the

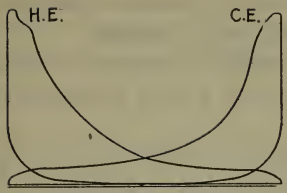


FIG. 18-a

FIG. 19-a

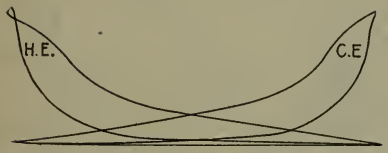


FIG. 20-a

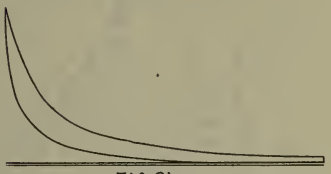


FIG. 21-a

DIAGRAMS FROM UNI-DIRECTIONAL FLOW ENGINE, SINGLE-ACTING ENGINE, LOCOMOTIVE AND DIESEL ENGINES

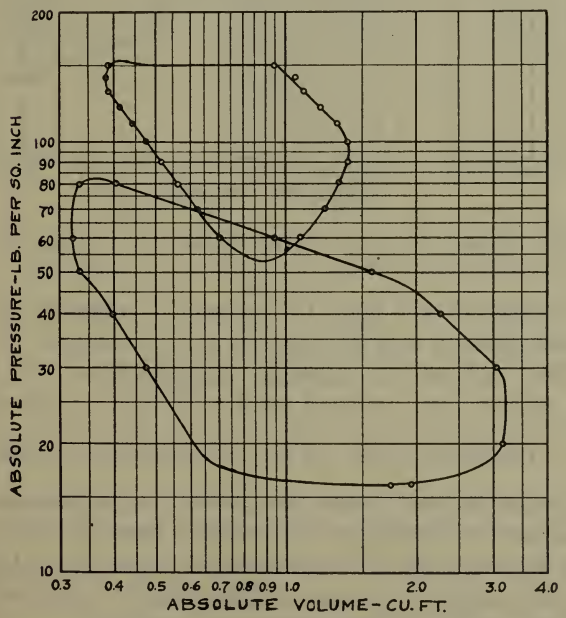


FIG. 19-b WESTINGHOUSE SINGLE-ACTING 13 IN. BY 22 IN. BY 13 IN. STEAM ENGINE

extra annual cost of fuel and water due to leakage equals the annual interest on the money necessary for general repairs.

123 It is apparent that the existing methods of determining when general repairs are necessary are in a very haphazard state as

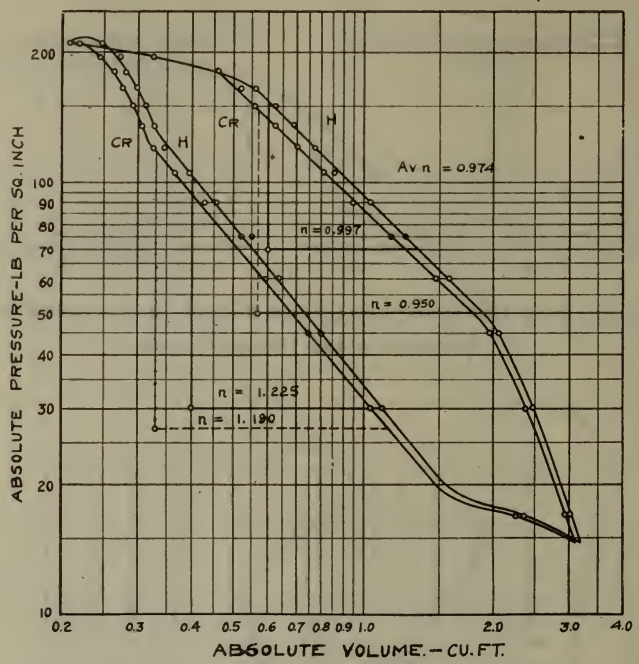


FIG. 20-b PURDUE 16 IN. BY 24 IN. SUPERHEATED STEAM LOCOMOTIVE

regards economy, and that the personal judgment of the man making the decision may be liable to great variation.

124 The method of detecting leakage from the logarithmic diagram offers a more rational solution of this important question.

INTERPRETATION OF THE LOGARITHMIC DIAGRAM

125 In the discussion upon the effect of wrong clearance, leakage, and excessive condensation [1] upon the lines of the logarithmic diagram, it was assumed, for the sake of clearness, that only one of these effects existed at one time. The examples were selected so as to illustrate only one of these effects in each case.

[1] Discussed in Appendix.

126 Cases occur where several of these effects exist at one time in the same diagram. The separation of one effect from another is not an exact process. However, the character of the curves showing

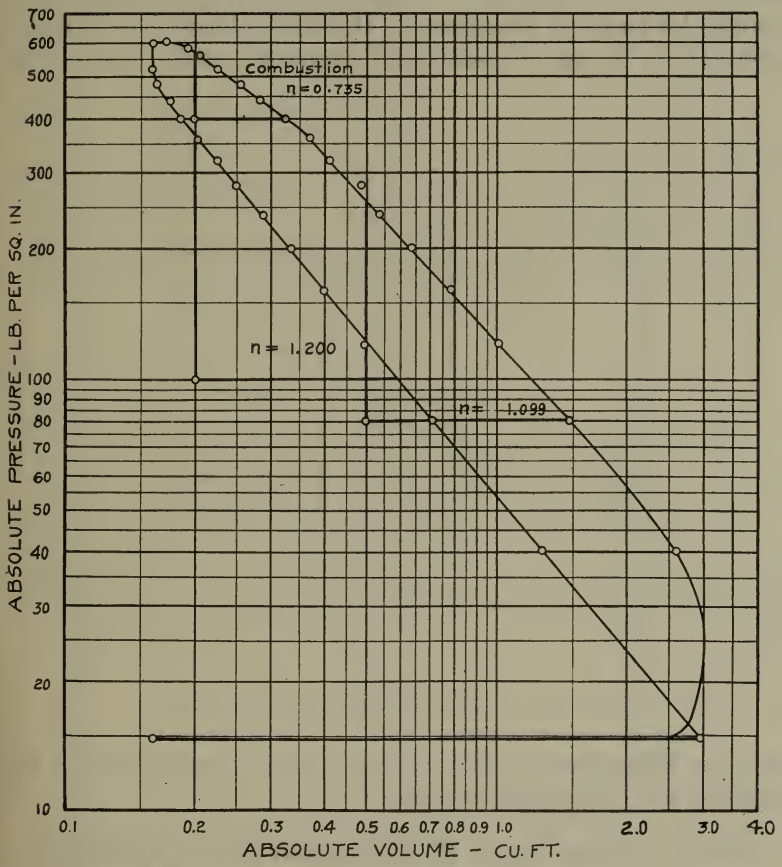


Fig. 21-b Diesel 16 in. by 24 in. Oil Engine, Crude Oil

excessive condensation, wrong clearance, and leakage are quite different. For instance, wrong clearance affects the lines throughout their length. Excessive condensation, in the cases of the steam diagrams examined, always affects only the upper parts of the curves. Leakage, as has been mentioned, affects materially only the upper and lower thirds of the lines, where these lines extend

from the initial pressure to nearly the back pressure. When excessive condensation and large leakage exist together, no close approximation of the clearance can be made.

127 An adequate treatment of the segregation of these various effects; when found together, is beyond the scope of this paper. The treatment is long and complicated. It has been found, however, that experience in the use of logarithmic diagrams enables one to sepa-

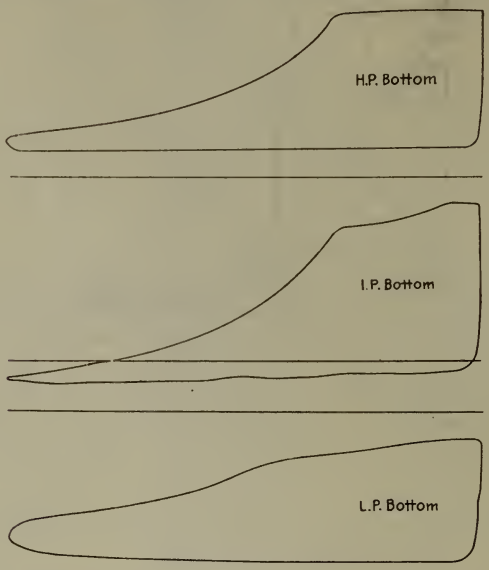


FIG. 22-a DIAGRAMS FROM PUMPING ENGINE

rate these effects qualitatively, in some cases, from the form of the expansion and compression curves.

COMMON ERRORS MADE IN ANALYZING STEAM INDICATOR DIAGRAMS

128 *The Use of the Equilateral Hyperbola as a Standard of Comparison.* The values of n for the expansion curves of steam indicator diagrams are not closely constant, but are subject to a very wide range of variation. The extent of this variation may be seen from the examples given in the logs of the tests made, and in the Appendix. The range of variation found in the present investigation is from 0.70 to 1.34.

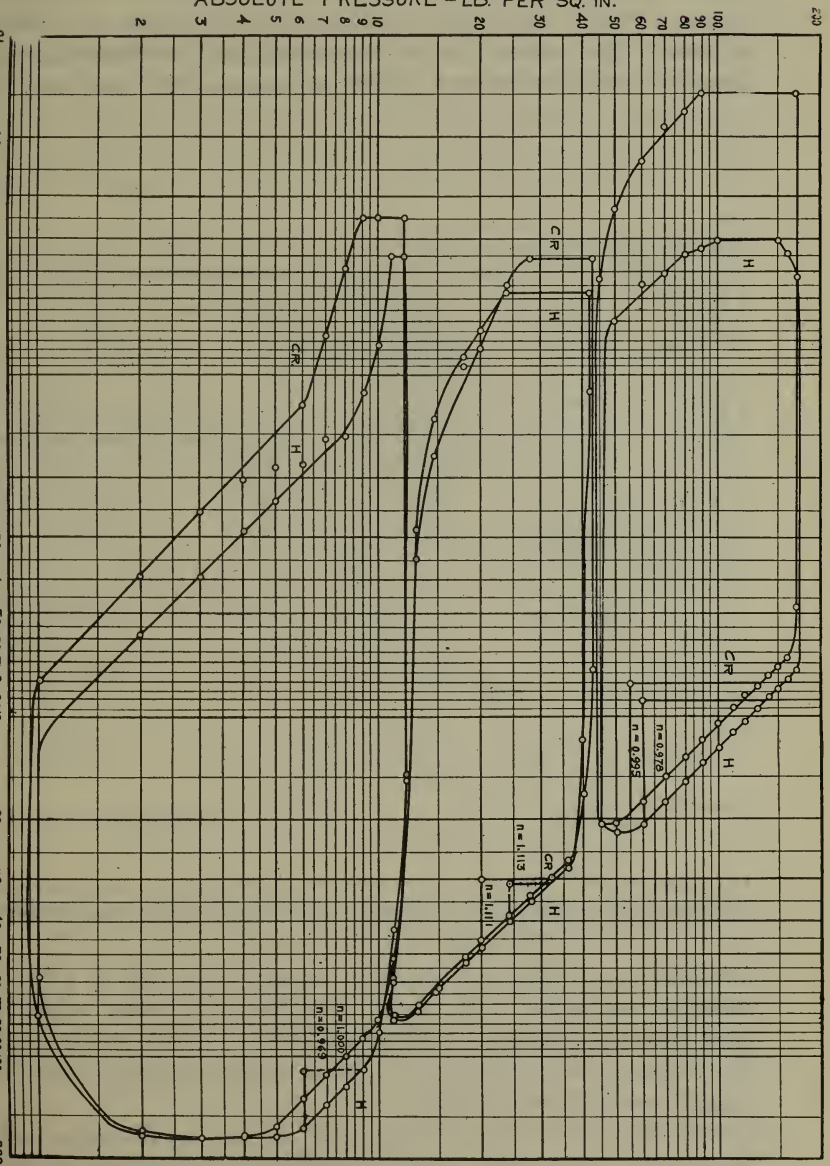


FIG. 22-b ALLIS-CHALMERS PUMPING ENGINE 28 IN. BY 54 IN. BY 80 IN. BY 60 IN. SATURATED STEAM

129 The range of values in the engine tested was from 0.835 to 1.234. The average values were 0.947 for the tests run with saturated steam, 1.056 for the tests run with superheated steam, and 1.004 for all tests. The values of n for most engines of ordinary size using saturated steam at normal cut-off is between 0.95 and 1.05, while

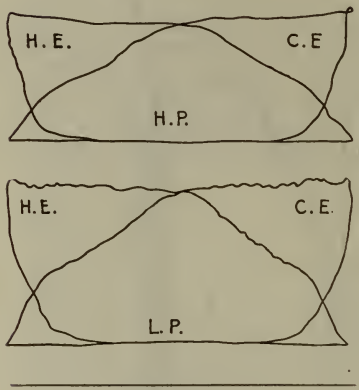


FIG. 23-a

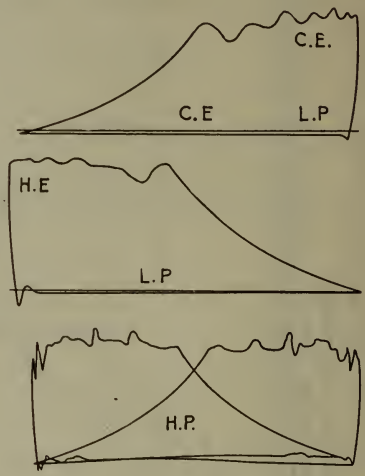


FIG. 24a

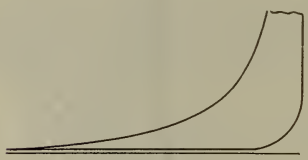


FIG. 25-a

FIG. 26

DIAGRAMS FROM COMPOUND AIR LOCOMOTIVE, TWO-STAGE AIR COMPRESSOR, SINGLE-ACTING AMMONIA COMPRESSOR AND CORLISS ENGINE WHEN STARTING

for superheated steam the range is usually from 1.00 to 1.30. For saturated steam the value of $n=1.0$ is about an average value. The only meaning that the average value of $n=1$ ever possessed is that the average value of X_c, in the class of engines examined, lies in the range between 0.60 and 0.70.

130 The law of Boyle or Mariotte, or the law of isothermal expansion of a perfect gas, has no bearing of any kind whatsoever on the expansion of steam in a cylinder. The equilateral hyperbola sometimes occurring in steam cylinders is only a special case of

expansion according to the polytropic law $PV^n = C$, while Boyle's law is another special case which never occurs in steam engine practice.

131 Because of the agreement in form between Boyle's law and the equilateral hyperbola (the special case of the law, $PV^n = C$, where $n = 1.0$), this latter curve has been called the ideal or theoretical curve of expansion to which curves in practice are supposed to approach as a measure of practical perfection in the use of steam. The

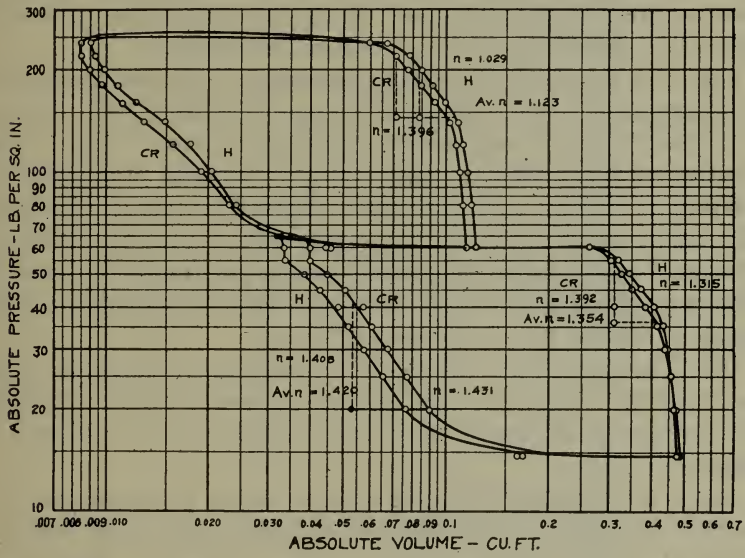


Fig. 23-b Porter Air Locomotive; Compound 5 in. by 10 in. by 10 in.

equilateral hyperbola is in no sense whatsoever an ideal or theoretical curve, and its use for the purposes of comparison is an empirical or arbitrary convention only. It should be called the conventional expansion. It has even been contended that because an expansion curve did not coincide with the equilateral hyperbola, some grave fault exists in the engine. A value of n may be as low as 0.60 with no graver fault than very excessive initial condensation, while a value of 1.35 may be found from no graver fault than that of using steam superheated about 250 deg. fahr.

132 The only rational use of applying the equilateral hyperbola to steam PV-diagrams is to act as a guide to see whether n is greater

or less than 1.0. If the actual curve is not close to this hyperbola, if no faults exist, and if the cylinder is non-jacketed, then this fact means that the value of X_e for the case examined is less than about 0.60 or greater than about 0.70. The assumption that $n=1.0$ as a standard of expansion is equivalent to assuming that the value of X_e is standard at about 0.65. No engineer would seriously propose that the value of X_e of 0.65 should be selected as a standard of economy.

133 The elaborate theory of analysis built on the assumption that $n=1.0$ is the natural result of the use of averages in any art where the actual facts have never been investigated.

134 The use of the equilateral hyperbola to predict the form of PV-diagrams for the purposes of design is satisfactory in the case of ordinary sizes of engines using saturated steam. When steam superheated over 100 deg. fahr. is used, the value of n should be assumed at between 1.10 and 1.25. The high values of n obtained with highly superheated steam in large engines alter materially the division of work and the tangential forces acting from those obtained when n is assumed to be 1.0. This fact should be considered in the design of engines to use superheated steam.

135 The use of the equilateral hyperbola to obtain the ratio known as the "diagram factor" has no rational basis, but its use for this purpose gives results which are valuable for the purpose of design.

136 *The Graphical Method of Approximating the Clearance.* If n has the value 1.0 on a PV-diagram, the clearance may be found by locating the zero of volume on the zero line of pressure. This process is performed graphically by reversing the method used in constructing the equilateral hyperbola.

137 In actual expansions, however, n is almost never exactly equal to 1.0, but is greater or less as already explained. The accuracy of the result by this method is dependent on how close the value of n approximated 1.0. The clearance obtained by this method may be as much as 100 per cent larger or smaller than the actual clearance volume in ordinary cases, while errors of 25 per cent and 50 per cent are very common. Where errors of this size are possible, the method is of no use for important work.

138 A rational method of approximating the clearance cannot be based upon the assumption that $n=1.0$, but only upon the fact that it be of a constant value, the value itself being immaterial.

139 *Combined Steam Indicator Diagrams.* Very little of value is obtained from the combined PV-diagrams of steam engines, except the measure of the diagram factor for the purpose of design.

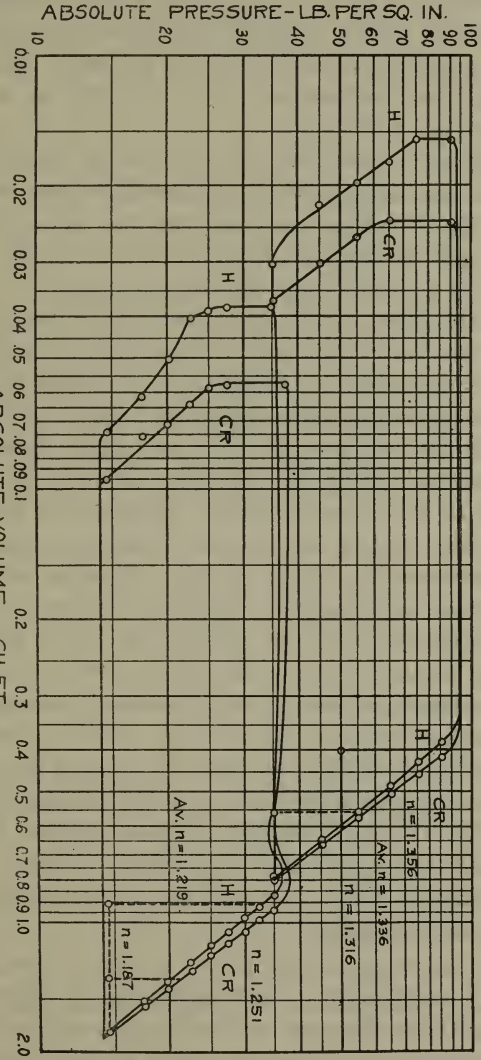


Fig. 24-b Ingersoll-Sergeant 12¼ in. by 18¼ in. by 12 in. Two-Stage Air Compressor

140 One of the uses that has been made of the combined diagram is to see whether continuity of expansion exists. It has been assumed by various writers that continuity of expansion should exist.

A study of the relations of X_e and n shows that continuity of expansion does not and should not exist except under very special conditions.

141 *Division of Feed for Applying Hirn's Analysis.* One of the requirements of Hirn's analysis is that we know exactly how much steam was admitted to each end of a cylinder. These amounts are not usually equal in practice, so an assumption must be made to cover the needs of the case.

142 The usual assumption is to divide the feed between each end of a cylinder in the ratio of the values of the mean effective pressure shown by the PV-diagrams from the two ends. This assumption is probably not far from the actual division in most cases, and is the best that can be done under the circumstances.

143 In the light of the facts presented in this investigation, this feed may now be divided on a more rational basis. It has been found that the presence of the piston rod in only one end of the cylinder has no appreciable effect upon the value of n. This fact enables us to divide the feed according to the volumes filled and the values of X_e as determined by the resulting values of n. This method is believed to be the closest solution obtainable in the case where the supply for each end of the cylinder cannot be separately measured.

144 *Computing the Weight of Steam Retained in Compression.* A carefully made investigation indicates,[1] when saturated steam is used, that the steam in the cylinder is dry or even very slightly superheated at the closure of the exhaust valve. As shown by the logarithmic diagrams, leakage of the steam in compression continues until the exhaust valve, in closing, has acquired considerable lap.

145 The point which is selected to compute the volume of steam retained in compression generally lies between the points A to B and C to D of Fig. 10. As the weight of steam retained is yet decreasing, the point selected nearly always accounts for more steam than was actually retained. In other words, we do not find the value of X_e to be as high as it actually is.

146 The following method shown in Fig. 10 has been adopted in the present investigation. The straight line of the compression curve on the logarithmic diagram, or the line of constant weight of steam mixture, is prolonged dotted as shown to the back pressure. The intersection of this prolonged line with the back pressure line ex-

[1] George Duchesne, Revue de mécanique, July 1899; quoted in Power, Jan 10, 1911, p. 71.

tended is taken as the volume of dry steam retained in compression. This method almost always gives less steam retained in compression

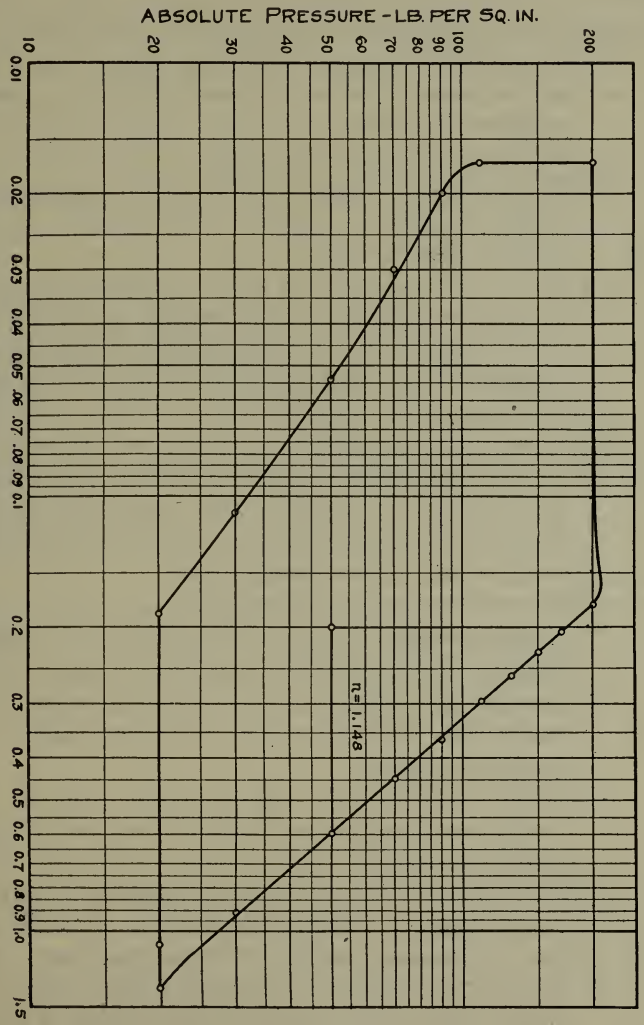


FIG. 25-b YORK SINGLE-ACTING 12½ IN. BY 18 IN. AMMONIA COMPRESSOR

than the ordinary method, and is believed to be rational in the present state of knowledge of this subject.

CONCLUSIONS

147 The following conclusions have been drawn from the results of the investigation relating to all elastic media, as applicable to the cylinders of reciprocating engines using elastic media and having, as a part of the working cycle, an expansion, a compression, or both:

a The indicator diagram, taken by means of a correct reducing motion and with a reliable indicator, contains the evidence necessary for a very complete and useful analysis of cylinder performance.

b The logarithmic diagram, derived from the indicator diagram, discloses a new and very complete analysis of cylinder performance.

c Free from the influences of leakage, wrong clearance, wrong location of the line of zero pressure, and excessively low speed (principally in steam engines), expansion or compression of an elastic medium takes place substantially according to the law, $PV^n = C$.

d The clearance of cylinders may be found by graphical trial in the logarithmic diagram to within 5 per cent to 10 per cent of the clearance volume, depending as the clearance itself varies from 20 per cent to 2 per cent, respectively, of the piston displacement.

e The cyclic events, even though entirely obscure in the indicator diagram, may be located on the logarithmic diagram (when plotted on logarithmic paper of 5 in. per square) to within $\frac{1}{16}$ in., this quantity being the equivalent of about $\frac{1}{32}$ in. when retransferred to the indicator diagram.

f Leakage (if appreciable) may be reliably detected from the logarithmic diagram, and may, in some cases, be approximated in volume.

g The weight of steam retained in compression should be obtained from the logarithmic diagram by prolonging the line of constant weight of steam mixture to the back pressure line extended; the intersection of these two extended lines is the volume of steam which is retained.

148 Acknowledgment is made to Prof. G. A. Goodenough, through whose kind offices permission was secured to undertake the investigation, and who at all times has given his hearty encouragement to the work; also, to Dean W. F. M. Goss, Prof. O. A. Leutwiler, C. M. Garland, Purdue University, and to the various firms which have furnished indicator diagrams and tests for analysis.

APPENDIX

PART 1 METHOD OF CONSTRUCTING THE LOGARITHMIC DIAGRAM

149 The logarithmic cross-section paper used in this investigation consists of four squares arranged two each way. These squares are 5 in. each way, making the four squares together 10 in. each way. The use of four squares enables values to be plotted ranging from 0.1 to 10.0, 1.0 to 100.0, etc., thus giving a range of ten times the values obtainable if only one square were used.

150 *Construction of the Logarithmic Diagram.* The coördinates of the *PV*-diagram are proportional to pressure and stroke, the latter being proportional to the volume displaced by the piston. The coördinates of several points on the *PV*-diagram are found in terms of absolute pressure, preferably in pounds per square inch, and absolute volume, preferably in cubic feet, these being the units used in steam tables.

151 The method of transferring the *PV*-diagram to the logarithmic form is described in detail for the diagrams of test 30, given in Fig. 9 (shown reduced in size). The method of drawing the pressure ordinates is shown on the crank-end card *ABYX*. Perpendiculars *QR* and *EX* are drawn to the atmospheric line *EQ*, and pass through the extreme stroke positions of the diagram. The distance *EQ* is then the length of the diagram. *OM* is laid off perpendicular to the atmospheric line *EQ* (extended) which was drawn by the indicator pencil. *OM* is the line of zero volume, and is drawn at a distance *FE* from the admission end *EX* of the diagram, the distance *FE* being the same length in per cent of the line *EQ*, or length of the diagram, as the proportion that the per cent of clearance, or waste space, of the cylinder bears to the piston displacement. In this case the length of the diagram is 3.99 in., and the clearance is 7.04 per cent. The length *FE* is therefore 0.0704 multiplied by 3.99, or 0.281 in. *ON* is the line of zero pressure and is drawn a distance *FO* below the atmospheric line, to the scale of the spring used in obtaining the *PV*-diagram. This distance is proportional to the barometer reading, corrected for temperature, prevailing at the day and place of the engine test. In this case the corrected barometer reading was 14.2 lb. per sq. in. absolute, so the distance *FO* is $\frac{14.2}{79.0}$ or 0.180 in.

152 From *ON* points are laid off on *QR* and *EX* corresponding to the absolute pressures at the intervals where it is desired to read off the corresponding volumes. Fine lines are drawn connecting similar pressure points, as 19.8–19.8, 29.1–29.1, etc. The volumes *G–A*, *G–B*, *H–D*, *H–C*, etc. are read off in hundredths of an inch to the nearest half hundredth. The tabular form used in this investigation is given in Table 6 for the diagrams of Fig. 9 taken in test 30. Thus the length *G–A* is read off as 0.66 in., and is given under the column for 19.8 lb. pressure, headed Compression, for the crank and diagram. The volumes in inches are then multiplied by the constant ratio which 1 in. of length of the diagram bears to the displacement of the piston. From Table 6 it is seen that the piston displacement of the crank end is 1.523 cu. ft., and the length of diagram 3.99 in., hence, the ratio is $\frac{1.523}{3.99}$ or 0.382 cu. ft. of piston displacement per inch

of diagram length. The length G–A in cu. ft. of displacement now becomes 0.66 multiplied by 0.382, or 0.252 cu. ft., the volume of steam present at this point. This process is repeated at intervals until the coördinates of from 10 to 30 points are determined. In the diagram shown in Fig. 9, the coördinates of 18 points were found in each diagram.

153 The coördinates of P and V are then plotted on logarithmic cross-

TABLE 6 CONSTRUCTION OF THE LOGARITHMIC DIAGRAMS OF TEST 30 †

Number	Head End					Crank End				
	Absolute Pressures, Lb. per Sq. In.	Volumes				Absolute Pressures, Lb. per Sq. In.	Volumes			
		In.		Cu. Ft.			In.		Cu. Ft.	
		Compression* X to A	Expansion* Y to B	Compression	Expansion		Compression	Expansion	Compression	Expansion
1	133.0	0.39	1.18	0.156	0.470	134.3	0.30	1.205	0.115	0.4605
2	113.4	0.31	1.36	0.124	0.5425	115.3	0.28	1.39	0.107	0.531
3	95.0	0.31	1.625	0.124	0.649	96.3	0.28	1.65	0.107	0.6305
4	76.3	0.31	2.02	0.124	0.8055	77.3	0.28	2.06	0.107	0.7875
5	57.1	0.31	2.705	0.124	1.080	58.1	0.295	2.76	0.113	1.055
6	38.9	0.375	4.03	0.150	1.610	38.7	0.385	4.12	0.147	1.572
7	29.4	0.48	4.22	0.192	1.685	29.1	0.485	4.26	0.185	1.626
8	19.5	0.70	4.26	0.279	1.699	19.8	0.66	4.20	0.252	1.600
9	15.0	1.10	3.06	0.439	1.220	15.0	0.93	3.77	0.355	1.440

* Letters refer to Fig. 9, Crank End.
† Final results given in Table 1.

	Head End	Crank End
Length of indicator diagram, in	3.95	3.99
Ratio of clearance to piston displacement, same end	0.0789	0.0704
Length on diagram proportional to clearance ratio, in	0.312	0.281
Length of diagram, plus clearance, in	4.26	4.27
Piston displacement (cylinder 12.02 in. by 24 in.), cu. ft	1.575	1.523
Clearance volume, cu. ft	0.124	0.107
Displacement plus clearance, total volume, cu. ft	1.699	1.630
Ratio, cu. ft. per in. of length on diagram, cu. ft	0.399	0.382
Scale of indicator spring per in. of ordinate, lb	77.0	79.0

section paper, as shown in Fig. 10, which are the logarithmic diagrams derived from the PV–diagrams of Fig. 9. The points plotted in Fig. 9 are taken from the columns headed Cu. Ft. at the pressures shown. A smooth curve is drawn through the points thus plotted, and the diagram is in shape to be studied.

154 *Examples of Logarithmic Diagrams from Various Types of Engines.* The logarithmic diagrams obtained by plotting the indicator diagrams as already described, are of a distinctly different form from either the PV–diagram or the temperature-entropy diagram. While the various typical forms of PV–diagrams assume rather different forms after plotting, yet the resultant figures retain in a general way the peculiar characteristics of each PV–type except that these peculiarities are exaggerated.

155 Various typical PV–diagrams are given in Figs. 16–25, which include examples from many types of engines using steam, gas, air and ammonia as the

active media. The values of n for the curves are given in each figure and it will be noted from these diagrams how closely the law $PV^n = C$ holds for the expansion and compression curves from a great variety of engines using different media.

156 It has long been known that adiabatic expansion, or compression, of any elastic medium takes place substantially according to this law with the different values of n for different media. In practice, however, expansion is never adiabatic, but is changed in character by the presence of the metal surrounding the working medium, and by imperfection of mechanism. The question then arises as to how this expansion has been changed in character and whether it still obeys the law $PV^n = C$.

157 Many investigators have examined the curves from actual diagrams to clear up this point for steam.[1] These include those who find that expansion in steam cylinders takes place substantially according to the law $PV^n = C$, but that n varies in value between wide limits in different cases.

158 An examination was made of the curves of 296 diagrams from the cylinders of 47 different engines using steam, gas, air and ammonia to investigate this point. As a result, it may be stated that, in the cases of the great majority of engines using elastic media, expansion and compression take place substantially according to the law $PV^n = C$. Certain exceptions, however, have been found, and the causes studied.

159 *Cases where the Law $PV^n = C$ does not hold.* The curves of expansion and compression, obtained from PV-diagrams, do not always follow the law $PV^n = C$. This fact is due to several causes, some of which have been definitely determined:

160 (*a*) *Wrong Clearance, or Wrong Location of the Zero Line of Pressure.* The law, $PV^n = C$ is only true where P and V are measured in absolute units. The clearance must be accurately determined in order to give absolute values of V. The scale of the spring, for the PV-diagram analyzed must be known, and also the atmospheric line drawn by the indicator, in order to locate the zero line of pressure. The units used for P or V may be of any denomination, but they must be measured from the zero of P and V.

161 When a curve $PV^n = C$ is plotted on logarithmic paper, the resultant curve is a straight line. The value of n as already explained, is the slope of this line measured from any two points. When the values of P and V are not absolute values, this curve is no longer a straight line, but becomes a curve of the second degree. The value of n being the slope obtained from two points on this curve, is no longer constant for all parts of the curve, but varies from point to point.

162 Therefore, when PV-diagrams are transformed to logarithmic diagrams, the values of both P and V must be measured in absolute units. When these values are not in absolute units, the resulting curve is not of the form $PV^n = C$, and therefore is not a straight line on logarithmic paper. The form of the curve obtained, when the values of V alone are not in absolute units, is given in Fig. 11-b.

[1]Technical Thermodynamics, Zeuner, vol. 2, p. 111.

Recherches experimentales, Leloutre, Bulletin de la Société industrielle du Nord de la France 1874.

Zur Theorie des Indikatordiagrammes, Luders, Zivilingenieur, 1881, vol. 27, p. 225.

The Steam Engine, Perry, p. 106.

163 (*b*) *Leakage.* The law $PV^n = C$ is applicable only in cases where the weight of the working medium remains practically constant during any expansion or compression. When this weight changes materially either by leakage into, or out from, the cylinder containing the medium the resulting curve no longer obeys the law, and it becomes a curve in the logarithmic diagram. This fact is very clearly shown in the curves of the logarithmic diagram derived from cylinders in which large leaks were known to exist. The examples showing this condition have already been discussed.

164 (*c*) *Low Speed in Steam Engines.* Very low speeds of rotation, together with very low piston speeds, will cause the compression curve to deviate from the law $PV^n = C$. The most common cases of this effect are seen in the "hook" or excessive condensation, near the upper end of compression curves. These hooks are found almost altogether in small engines having very low piston speeds, and in larger engines with small clearance, having very low rotational speeds, as in pumping engines.

165 Diagrams containing hooks in the compression curves have been published by Professor Dwelshauvers-Dery[1] from the experimental engine at Liege. This is an example of a very low speed in a small engine. The size was 12 in. by 24 in. and the speed from 30 to 60 r.p.m. That this hook was caused by low speed and consequent excessive condensation, was proved in a measure in the case of the engine used by the author. This engine was also 12 in. by 24 in. but was operated at from 90 to 150 r.p.m. In no case was a hook in the compression curve obtained during the tests, although some 1600 diagrams were taken. On one occasion, however, the diagram shown in Fig. 26-a was obtained. This was taken just after the engine was started from cold, and had been brought up to a speed of 120 r.p.m. In conjunction with the other diagrams obtained from this engine in regular operation, this hook is believed to be due to excessive condensation while the cylinder was comparatively cold.

166 The three causes just treated are believed to be the important conditions that cause the curves in practice to depart materially from the law $PV^n = C$. Sometimes only one of these conditions is present, while in other cases, a combination of these may influence the resulting curves. The separation of these conditions by their effect on the curves has already been treated.

PART 2 PRINCIPAL DIMENSIONS OF ENGINE USED IN TESTS

a Type—Horizontal single-cylinder, double-eccentric, non-condensing, variable speed, heavy-duty frame, Reynolds Corliss engine

b Class—Belt drive for mill work

c Maker—Allis-Chalmers Company, Milwaukee, Wis.

d Rated Power of Engine—100 h.p. at 115 lb. initial pressure above atmosphere on indicator diagram, $\frac{1}{4}$ cut-off, and 120 r.p.m.

e Cylinder dimensions

a	Bore (measured while hot), in	12.02
b	Stroke, in	24.00
c	Diameter of piston rod, in	$2\frac{3}{16}$

[1]Power, June 28, 1910, p. 1165.

f Clearance—In per cent of volume displaced by piston per stroke
  a Head end 7.89
  b Crank end 7.04
g Speed—Controlled by fly-ball governor with variable gear ratio between main shaft and governor, giving any engine speed from 20 to 160 r. p. m. Usual speed 120 r.p.m.

167 The cylinder of the engine is not steam-jacketed on the ends but is partly jacketed on the barrel by the steam chest, the latter covering about one-sixth of the barrel surface. The exhaust passages are separated from the lower part of the cylinder barrel by a dead air space formed in the cylinder casting.

DISCUSSION

Arthur L. Rice said that Mr. Clayton's paper gives the engineer for the first time a chance to keep track closely of what the engine is doing at all times while it is in actual operation. He had had occasion to use this method since the publication of the paper in The Journal. In the use of a high pressure of steam in the engine and a high speed, where the indicator card is a small one, it is absolutely essential that very great accuracy should be observed; otherwise the indications as to the amount of steam consumption may be quite misleading. As to the percentage of accuracy of this method as compared with weighing the steam from the surface condenser, Mr. Rice found from tests that the variation was from 2 to 6 per cent. The logarithmic curve may be plotted directly in inches, without reducing the volume to cubic feet. The only volume of interest is that at cut-off, which is a single volume and easily computed. It is, however, necessary in all cases to take care that the logarithmic paper used is accurate, which often is not the case with the paper in the market.

R. C. Stevens disagreed with Mr. Clayton's treatment of the problem of leakage, and questioned where the author takes into account the leakage of steam directly out of the exhaust, or that part of the steam which enters the steam chest but, by reason of the valve leakage, passes directly out of the exhaust without entering the cylinder. That this loss is enormous, even in many new engines, is too little appreciated. For example, in a case quoted by Mr. Stevens from an article by George Mitchell in Power, October 11, 1910, it was as high as 22 per cent in an engine specially fitted by its makers with a new valve. What is true of piston valves is doubly true of the flat balanced valve with pressure plate. Tests on one such engine of 80 h.p. capacity which the owners considered in good condition showed 624 lb. steam leakage per hour, or about 21 h.p. wasted, or over 25 per cent leakage. It is common practice to allow 0.003 to 0.004 in. freedom between the valve and pressure plate when newly fitted, and in operation the valve wears small and the seat wears large. Altogether, valve leakage is far more important than piston leakage, which may be almost negligible for years. Mr. Stevens therefore challenged Mr. Clayton's assertion that steam con-

sumption may be approximated from indicator diagrams to within an average difference of less than 4 per cent from the test results, and only in rare cases as much as 8 per cent. For, if 20 per cent or more of the steam that enters the throttle goes out of the exhaust without making its presence felt in any way on the indicator, it is difficult to understand how any analysis of the indicator diagram or its derived logarithmic diagram can establish the actual steam consumption of an engine to within even 20 per cent of accuracy.

Sanford A. Moss pronounced the paper to be the first new thing on steam engines for many years, but insisted on the necessity of taking leakage into account. Steam engines are seriously hampered by leakage and when this occurs Mr. Clayton's method cannot be entirely depended upon, unless suitably modified. In the paper an attempt is made to give a variation of the law with pressure, and it did not seem to him that the tests warrant any distinction between different pressures and that the author should use the equation from the average of all pressures rather than the chart of Fig. 6.

R. C. H. Heck. In the use of the logarithmic diagram as a means of investigation, in the proposition that exponent n is practically constant for the respective curves of a particular steam diagram, and in establishing a definite relation between n and quality at cut-off, this paper makes three distinct contributions to the field of research and knowledge which lies between the performance of the actual engine and that of the ideal engine with a non-conducting cylinder. In this region, rational deduction from simple premises is not available, chiefly because it is not possible to simplify and make definite the needed premises: consequently, theory can be built up only by such formulation of empirical results as is here undertaken.

To say that n varies widely with different conditions of working is merely to give special form to a statement of long recognized fact. We know that the approximation of actual expansion and compression curves to the equation $PV = C$ is very rough. A common range of variation with saturated steam is from 5 to 10 per cent, while in extreme cases the product PV departs a good deal farther from constancy. Nevertheless, the hyperbola has been and will continue to be a useful reference

curve, not from any sanction in theory, but simply because it represents constant PV.

Critical judgment upon the method derived and proposed for estimating actual steam consumption could be given only after extensive trial and use. The showing in Table 4 seems to be very favorable. Manifestly, the scheme is as yet incomplete, since the service chart, Fig. 6, is limited in application to one exhaust pressure. In working along the older line of a formula containing size, speed, cut-off ratio, and range of steam tempera-

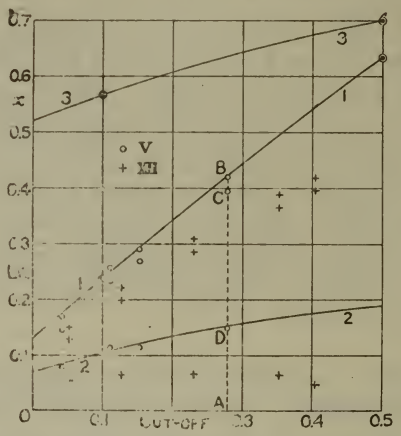


FIG. 26 PLOT OF STEAM QUANTITIES PER REVOLUTION ON CUT-OFF AS BASE, FROM TEST SERIES V AND XII, TABLES 1 AND 2

CURVE 1, ORDINATE AB, TOTAL STEAM PRESENT DURING EXPANSION, MADE UP OF MEASURED CONSUMPTION AC PLUS CLEARANCE STEAM CB.

CURVE 2, ORDINATE AD, MISSING STEAM AT CUT-OFF, FOUND BY SUBTRACTING INDICATED STEAM BD FROM TOTAL STEAM AB.

CURVE 3, CUT-OFF QUALITY x_c, CORRESPONDING WITH CURVES 1 AND 2.

CURVES ARE DRAWN FOR THE SATURATED STEAM TESTS ONLY; EXTENSIONS TOWARD THE RIGHT ARE GUIDED BY THE FORM OF OTHER SERIES, WHICH HAVE LONGER RANGES OF CUT-OFF; THE CIRCLED DOTS ON ORDINATES AT 0.1 AND 0.5 CUT-OFF SHOW VALUES USED IN THE EXAMPLES; THE SUPERHEAT TESTS, SERIES XII, ARE HERE GIVEN FOR PURPOSES OF COMPARISON ONLY.

ture as elements, the writer has consistently found the last the most difficult to represent satisfactorily.

It is a matter of considerable interest to know just what sort of thermal interactions are involved in the exact production of the curve $PV^n=C$ with a steam and water mixture. General relations cannot be expressed in serviceable shape, therefore it is necessary to resort to the solution of numerical examples. Taking typical conditions determined by test series V, Table 1, as

plotted in Fig. 26, the writer has calculated results which are presented graphically in Figs. 28 and 29.

The controlling conditions for this pair of examples are as follows:

General:		
Pressure at cut-off, lb. absolute	111	
Corresponding steam temperature, deg. fahr	335.5	
Piston displacement per revolution, cu. ft	3.10	
Clearance, per cent	7.5	
Particular:	Case *A*	Case *B*
Quality at cut-off, *Xc*	0.5667	0.7000
Exponent *n*	0.92	1.025
Weight of steam mixture, lb	0.2375	0.6332

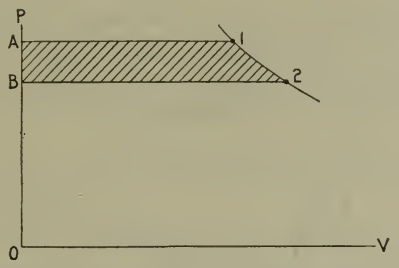


FIG. 27 DIAGRAM SHOWING METHOD OF CALCULATING RESULTS IN FIGS. 28 AND 29

Thermal quantities were first worked out for the weights of steam just named, then were reduced to b.t.u. per cu. ft. of piston displacement for use in the diagrams.

The method of calculation is illustrated by Fig. 27 and the relations involved are expressed by the equation

$$h_1 + q = A \int_1^2 v dp + h_2 \quad \ldots\ldots\ldots\ldots [4]$$

Here h_1 and h_2 are heat contents or total heats at the beginning and end respectively of the expansion interval, or at points 1 and 2; q is the heat imparted to steam by cylinder walls; and

$$A \int_1^2 v dp = \frac{144n}{778(n-1)} (p_1 v_1 - p_2 v_2) \quad \ldots\ldots\ldots [5]$$

is the heat equivalent of the work done in expansion from p_1 *to* p_2, represented by area $A12B$. The scheme is to use temperature intervals, of 15 deg., passing from t to p through the steam table.

In Fig. 28, distances from the base to line 1 represent heat used in the performance of external work, calculated by equation [5]. Distances from 1 to 2 represent changes $(h_2 - h_1)$ in heat content, positive in case A where h_2 is greater than h_1 $(n =$

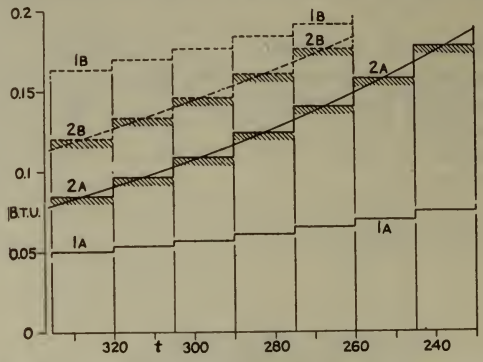


FIG. 28 RATES OF HEAT RECEPTION PER DEGREE OF CHANGE OF STEAM TEMPERATURE

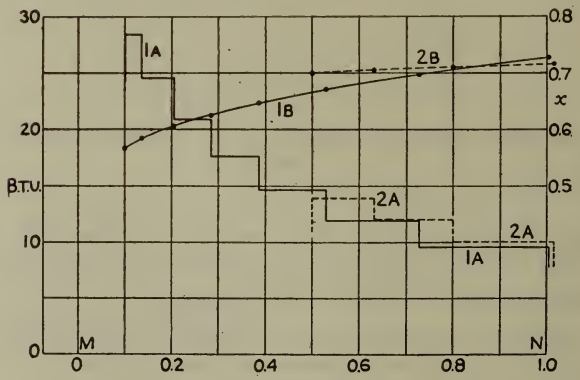


FIG. 29 RATES OF HEAT RECEPTION PER CUBIC FOOT OF PISTON DISPLACEMENT, ALSO CURVES OF STEAM QUALITY

0.92), negative in case B where h decreases with expansion $(n =$ 1.025). The net or resultant distance from base to line 2 shows heat received q.

Strictly, the areas beneath the several sections of the broken lines in Fig. 28 represent the heat quantities just named, while to the scale of the diagram the ordinates show rates of heat conversion, change, or transfer per degree of change of steam tem-

perature. Thus in case A, with 0.2375 lb. of steam, the heat added from 305 to 290 deg. is 5.05 b.t.u. This is divided by 3.10 to make it correspond with the steam in action per cubic foot of piston displacement and by 15 to make it heat per degree. In case B, with 0.6332 lb. of steam, the similar heat quantity is 6.76 b.t.u., to be reduced to the terms of the diagram by the same division. Because thus related to steam quantities other than 1 lb., the ordinates of curves 2A and 2B are not identical with specific heat; but they are proportional to unit heat capacity.

Of course, for this operation of hyperadiabatic expansion, the specific heats represented by curves 2 of Fig. 28 are negative. But the point of special interest here brought out is their increase with fall of temperature.

In Fig. 29 base MN represents 1 cu. ft. of piston displacement, with clearance OM added at the left; and it may also be taken to represent stroke of piston. The principal ordinate added is now, heat added divided by change of volume in expansion. Thus for the interval from 305 to 290 deg. in case A, heat q or 5.05 b.t.u. is divided by a volume change of 0.2417 cu. ft., and the quotient 20.9 is an ordinate of line 1A.

As bearing on the maintenance or production of a constant exponent n, we note that for case A, with early cut-off, a very rapid heat reception is required at first; but with later cut-off the requirement is more nearly uniform. Now the store of latent heat in the steam is relatively so large that a very small change in the manner of variation of quality x will set free or absorb quantities of heat which are of the same order of magnitude as those calculated for Fig. 28. We may therefore conclude that there is nothing in the showing of Fig. 29 to antagonize the empirical fact that for ordinary conditions of working n is practically constant. But we can see that for very early cut-off heat is likely to be less rapidly transferred at first than is required to produce the average value of n, while later the requirement may be exceeded. Such an action, with n at first large and pv decreasing, followed by decrease of n and increase of pv, is just what is observed in this case.

A few classic examples from our records are set forth in Fig. 30. The first two curves show the smooth form of the polytropic in the scheme of coördinates which is used. No. 3 is a case of very low speed, with rapid growth of pv at low pressure and accompanying decrease of n. No. 4 shows the opposite ex-

treme of high superheat, with fair approximation to a value of n considerably greater than unity. In the engine of No. 5 release was late, so that the expansion curve ran clear to the end of the stroke; and in this period of very slow piston movement there is time for a large inflow of heat, strengthened by jacket action. Curve 5 shows, then, condensation continuing past cut-off and very marked reëvaporation at the end of expansion. It

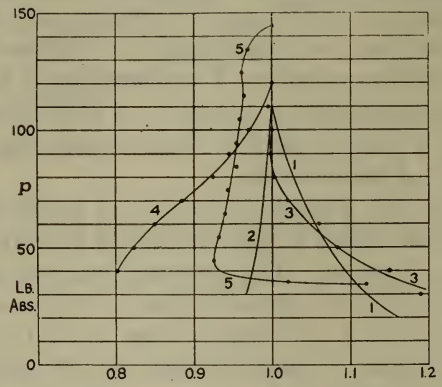


FIG. 30 PLOTS OF pv ON p, REFERRED TO THE INITIAL VALUE OF pv AT CUT-OFF AS UNITY

CURVES 1 AND 2, FROM EXAMPLES WORKED AND ILLUSTRATED IN FIGS. 28 AND 29 VALUES OF n=0.92 AND 1.025, RESPECTIVELY.

CURVE 3, TRANSACTIONS, VOL. 10, PAGE 742, FIG. 170, CYLINDER 17 IN. BY 30 IN., SATURATED STEAM, SPEED 8.64 R.P.M.

CURVE 4, TRANSACTIONS, VOL. 25, PAGE 267, CYLINDER 16 IN. BY 42 IN., SUPERHEAT 352 DEG. AT THROTTLE, 187 DEG. IN CYLINDER AT CUT-OFF, SPEED 103 R.P.M.

CURVE 5, TRANSACTIONS, VOL. 16, PAGE 176, LEAVITT PUMPING ENGINE AT LOUISVILLE, CYLINDER 27 IN. BY 120 IN., SATURATED STEAM, FULL JACKETS, SPEED 18.6 R.P.M.

is a striking example of how decidedly steam may fail to conform to the polytropic equation.

F. W. MARQUIS discussed the method of Mr. Clayton as a valuable aid in connection with the study of locomotive performance. It has heretofore not been possible to determine the steam consumption of locomotive engines under certain predetermined conditions of speed, cut-off, etc., by means of road tests, since in road service these conditions cannot be maintained constant for a long enough period to constitute a test. Mr. Clayton's method gives a way to determine the steam consumption under any conditions of speed, cut-off, etc., at which a locomotive can be oper-

ated, by means of indicator cards which can be taken while the locomotive is being used in regular road service, and a study of the effects of speed, cut-off, steam distribution, etc., can now be made by means of data obtained on the road tests. Further, from the logarithmic indicator diagram it is possible to determine accurately the times at which the different events of the stroke take place. This furnishes a comparatively simple method for determining the spring in locomotive valve gears. A study of this spring and its effect upon steam distribution, the stresses in valve gear parts, steam consumption, etc., with different types of valve gears, and under different conditions of running, is sure to prove of great value. The same method may also help to determine exactly why it is harder to make schedule time with passenger trains in cold weather than in warm.

Train resistance is greater in cool than in warm weather; a locomotive cannot make as much steam in cold weather, and again, steam must be furnished for heating the train. How much steam does it take to heat a train in cold weather? The steam thus used does not pass through the nozzle and help produce draft to burn the coal, as the steam used by the cylinders does. Is the amount large enough to affect seriously the steaming qualities of the locomotive? Information upon the above questions appears to be very meager, and has not in the past been easy to obtain. Now, however, it is possible to determine from indicator diagrams how much steam is used by the cylinders while the total steam used may be determined by means of tank measurements. The difference will be the steam used in heating and lighting the train, in running the air pump, etc.

C. D. Young. Good use has been made of the fact which has been established for some time that a practically constant quantity is obtained in the product of the absolute pressure by the nth power of the volume of steam under compression or expansion, the volume including the clearance space. This law has been well known mathematically as a straight line when expressed by its logarithmic equation, but Mr. Clayton has succeeded in defining what n stands for, showing that it is a quantity definitely related to the quality of the steam, and for any given engine working under given conditions the value of n depends on the quality of the steam.

The writer assumed that this followed Mr. Clayton's discovery

that the expansion and compression curves of indicator diagrams do as a general rule develop straight lines when absolute volumes are plotted against absolute pressures on logarithmic paper; and that where the plotted lines are not straight some defect in the cylinder is indicated and the amount of the discrepancy becomes an indication to some extent of the magnitude of the defect.

The writer feels that Mr. Clayton's discovery will develop to a point where it will be exceedingly valuable as a short-cut method and he is to be congratulated upon opening up a new field for study of cylinder performance of reciprocating engines. There are a number of statements in the paper, however, which he is inclined to question. For example, in Par. 8 the broad statement is made that since working with the new method of analyses existing methods have been found to be in a very crude state and many deductions from them are without foundation or meaning. He finds nothing in Mr. Clayton's paper to substantiate such a statement and would hesitate to criticize broadly the past work on engine performance.

Again in Par. 11 statements are made to the effect that facts as set forth were unknown previous to this method of investigation; whereas the writer believes that all the statements have been confirmed by work of others on this same subject.

The statement hardly seems warranted in Par. 74*c*, in which Mr. Clayton says that this method is more accurate than the average test, and is the only accurate method available for testing certain classes of engines. Assuming that the cards have been taken correctly and represent the actual cylinder performance, and before it is definitely determined what the actual water rate of cylinder is, the steam, either entering or leaving the cylinder should be measured by such means as are now available.

From the data obtained in Mr. Clayton's tests it is quite evident that the quality of steam mixture at cut-off when using superheated steam is in a large number of cases below the condition of dry and saturated steam and in Table 2, column 13, the quality of steam is given as below unity. The writer hesitates to criticize this figure, although he possesses data which would indicate that the quality at cut-off as given by him for superheated steam will not be as low as that which he has given and obtained graphically.

Until Mr. Clayton's method can be checked against some of the accurate tests which are daily being conducted at the Penn-

sylvania Railroad Company's Locomotive Testing Plant at Altoona, where all conditions can be controlled and accurate measurements taken, the writer hesitates to suggest further errors in the data as presented.

J. B. Stanwood. This paper is a good example of what an advanced scientific treatment of a well-known subject may develop. While the logarithmic treatment of expansion curves is not new,[1] the author has discovered that a relation of a very simple character exists between the quality of steam at cut-off in a steam engine and the expansion curve which develops from this quality of steam. This is an interesting fact which may prove of considerable value in further analysis of the action of steam in the cylinders of reciprocating engines.

In regard to the actual deductions drawn by the author, Mr. Stanwood calls attention to the fact that they are based almost entirely upon experiments made with a 12 by 24-in. Corliss engine, and believes that the applicability of his method to the determination of the actual values of initial quality of the steam can be determined only after comparing a number of experiments of the same character on different sizes of engines. In the instance at hand, leaving leakage out of account, it would seem that the initial quality of steam at cut-off is dependent upon two things: the amount of steam actually condensed up to cut-off, and the quality of steam in the clearance space just before the admission. If the quality is based simply on the difference between steam as accounted for by the indicator at cut-off, and the water consumption as determined by the amount condensed in the condenser, he fears the result is not correct, as no consideration has been taken of the quality of the "cushion" steam. It may be possible that the quality of cushion steam may be as low as 10 per cent, especially if Mr. Willan's theory is true that water retained on the clearance surface forms an important factor in creating cylinder condensation. Professor Zenner has stated that if he knew absolutely the quality of steam at any point during the compression he could determine the steam consumption of a steam engine from its indicator card

[1] See Prof. Perry in the discussion of Mr. Willans' paper on Non-Condensing Engine Trials; Proceedings of the Institute of Civil Engineers of Great Britain, vol. 43, 1888.

(leakage being neglected). This question of the quality of cushion steam may have something to do with the variations in the results obtained by Mr. Clayton in his comparison of different engines, as shown in Table 4.

There are probably few, if any, engines operating without leakage and the writer thinks there must have been quite an appreciable amount in the 12-in. by 30-in. engine tested by Mr. Clayton. For instance, in Table 1, with 109 lb. absolute pressure at the throttle and 25 per cent cut-off, this engine is stated to have a quality of steam of 58.8 per cent. This means a cylinder condensation of 41.2 per cent, which is high for this cut-off in comparison with most tests that have been made on Corliss engines. George H. Barrus in his book on Steam Engine Tests gives the result of his experience with these engines and he estimates as a result that the amount of condensation for 25 per cent cut-off is on an average about 24 per cent. If therefore there was no excessive leakage in this engine, then these experiments show a remarkable difference between condensation in small engines and condensation in larger engines of the same general type. In Table 4 there is recorded a test (Analysis No. 107) of a 16-in. by 42-in. Corliss engine, in which the quality of steam is 79.7 per cent. In the same table, with a Corliss cylinder 23 in. by 60 in., the quality of steam is 85.3 per cent. The writer suspects that these engines were among those tested by Mr. Barrus in his work entitled Steam Engine Tests, and if so, the cut-offs were about 25 to 30 per cent. Is there this difference between engines of 12 in. and 16 in. and 23 in. bore? If not, there must have been some other cause which would make the quality of steam so low in the 12-in. by 24-in. engine and the writer suspects that it is leakage.

The discovery of Mr. Clayton may lead to the solution of some problems heretofore unsolved, such as determination of the actual quality of steam in the compression spaces; prediction of steam consumption, especially as influenced by the size of cylinders; effect of clearance and clearance spaces on the economy of the steam engines. Designers may also be able to improve the proportion and construction of steam cylinders and valve gears by the character of the expansion curve under different conditions analyzed by this logarithmic method. The writer fears that a determination of the steam consumption of many types of

engines by this method would not be very reliable. There are so many cylinders in which the steam is short circuited directly from the steam side of the cylinder to the exhaust side of the cylinder that no evidence of this loss appears on the diagrams, so that the use of Mr. Clayton's method would not be at all satisfactory. There are many single valve engines with balanced valves in use in this country in which the leakage begins to increase from the day they go into service; any diagram taken from them without absolute knowledge of the valve conditions would be useless, and we would caution engineers as to the unreliability of this method for engines of this class. When valves are made tight and can be kept tight, the investigation along these lines may be helpful.

William D. Ennis was able in one or two instances to verify Mr. Clayton's law. A small automatic engine gave n=0.96, with a dryness at cut-off of 0.62. A Corliss engine gave n=1.02 average corresponding with an indicated steam rate of 26 lb. per i.h.p-hr., noncondensing. A study of values of n for the compression curve is interesting. Values exceeding 1.0 have repeatedly been found, along with such absence of high final temperatures as demonstrates that the customary Hirn assumption of initial dryness equal to 1.0 must be incorrect. Professor Ennis had found in two cases, with small automatic engines, n for compression=1.85 and 1.80 and in another instance it was apparently around 4.0.

From the thermodynamic side, the paper is of particular importance from the new light it throws on the question of heat interchange. Textbooks state that during the early part of expansion the steam continues to give up heat to the cylinder walls; afterward, as its temperature falls, again receiving this heat. But if the expansion line of the steam diagram conforms to the polytropic $PV^n=C$ there can be no such reversed transfer: the steam must be either steadily gaining or steadily losing heat. And, according to Mr. Clayton, it loses or gains according as it is dry or wet at the point of cut-off. In the light of the present paper, Professor Ennis is inclined to think that the apparent reversal of transfer has been generally due to leakage. In the Callendar and Nicholson experiments the steam temperature dropped to that of the metal of the cylinder head in

1/18 sec. after cut-off. Considering the lag and dampening of wall temperature fluctuations, the conditions thus favor a gain of heat by the steam practically throughout expansion. Strangely enough, also, the wall temperature taken at a point in the side wall just inside the face of the head, began to fall while the walls were about 17 deg. fahr. cooler than the steam.

The subject of steam leakage is one of the most important which the paper suggests. Leakage between two chambers will, other factors being constant, be proportional to the pressure difference and inversely proportional to the seal of the valve. It is most likely to occur:

a through the *steam* valve, just after cut-off, or just before admission

b through the *exhaust* valve, during the early part of a stroke

c through the *piston*, during the early part of a stroke.

In cases (*b*) and (*c*) the diagrams are scarcely likely to show leakage until after cut-off has occurred. Leakage through the steam valve, occurring just after cut-off, may be shown by a steam chest diagram. It raises the expansion curve during the early part of expansion. When it ceases, the true polytropic curve begins. Pre-admission leakage at this point raises the compression curve, and may take place at such a rate as to give a curve apparently polytropic, with an extremely high n value. A steam chest diagram may confirm the diagnosis. Leakage through either piston or exhaust valve, occurring just after cut-off, *depresses* the expansion curve. If it is the piston that leaks, a corresponding *rise* of pressure may in some cases be noted on the exhaust line of the compression diagram for the other end of the cylinder. If such leakage occurs before cut-off, the diagram in question will not show it, but the companion diagram may show a rapid rise of pressure during the early part of its exhaust period. Mr. Clayton's paper is thus far but a beginning. The bearing of the new law on the application of Hirn's analysis and the temperature-entropy diagram to actual tests will compel us to revise our judgment as to the utility of these forms of investigation. A new and undoubtedly fruitful field has been opened in steam engineering.

FORREST E. CARDULLO. It has long been known that the expansion of the working fluid in practically all reciprocating engines approximates the law $PV^n = C$, in which P is the absolute pressure, V is the absolute volume of the working fluid, and C is a constant. The paper before us proposes a most interesting method of cylinder analysis which takes advantage of this fact.

With that part of the paper which deals with the analysis of gas engines, air compressors and other reciprocating engines employing permanent gases, the writer is in complete accord. There are, however, many points in connection with that portion of the paper which deals with steam engines with which he cannot agree.

One of the first questions which arises in this connection is how closely does the expression $PV^n = C$, represent the actual facts in the case. Taking the adiabatic expansion of steam and plotting the relation between the pressure and the volume on logarithmic paper, we find that the expansion curve is convex toward the origin, but that the curvature is so very slight as hardly to be noticeable. When plotted to the scale which Mr. Clayton uses and for the range in pressures employed in reciprocating engines, the adiabatic expansion line is sensibly straight.

In Par. 164, Mr. Clayton states that the only irregularities observed in the compression lines of steam cards occur with low speed, and are in the form of a "hook" which occurs at the point where the saturation temperature of the cushion steam rises above the temperature of the cylinder walls. The writer has seen such hooks due to cylinder condensation, in the compression curves of cards from a $9\frac{1}{4}$ by 12 straight line engine running at 300 r.p.m. These hooks will always occur with any type of engine when the compression is sufficiently great. They are more pronounced in engines having a small clearance volume, and the higher the rotational speed of the engine the higher the compression pressure at which they occur.

During the expansion the steam is condensed at the beginning of the expansion line unless the drop in pressure during admission is so great that the saturation temperature of the steam falls below the temperature of the walls before expansion commences. Later on, instead of condensation there is re-evaporation during the expansion period. If the adiabatic line is straight on the logarithmic plane, it follows that the actual expansion line cannot be, but will have a decided bend in it at the point where the saturation temperature of the steam is equal to that of the cylinder walls.

This effect is well shown in the temperature-entropy diagram of many steam engines which has the form shown in Fig. 31. In this figure, at the point marked A, the expansion curve is tangent to the adiabatic, and this is the point where the temperature of the steam becomes equal to the temperature of the cylinder walls. The temperature of the cylinder walls rises with the speed of the engine and consequently the bend in the logarithmic expansion curve will rise, until in the case of many high-speed engines it may be higher than

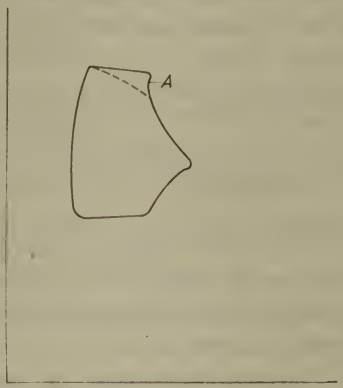


FIG. 31 TEMPERATURE-ENTROPY DIAGRAM SHOWING FORM OF EXPANSION LINE

the actual expansion line, which would be the case in Fig. 31 if the steam line was the dotted line instead of the full line.

The appearance of this bend is exactly the same as the appearance which Mr. Clayton attributes to leakage, and it may be questioned therefore whether leakage always exists when the expansion line has the form shown in Fig. 14-b.

In the case of a fluid expanding in a conducting cylinder, n is never a constant, but is a variable, although the variation is usually very slight and nearly uniform. Any appreciable irregularity in the logarithmic expansion or compression line is therefore evidence of leakage or of some other disturbing condition. We may therefore expect that a further study of the logarithmic expansion and compression lines will make clear some very interesting facts in connection with reciprocating engines.

Par. 42 states that the skin surface of the cylinder must be heated once every cycle from the temperature acquired by contact with the

exhaust steam nearly up to the temperature of the admission steam. This statement is misleading since the range in temperature of the metallic surface is known to be comparatively small, being only from a fifth or a tenth of the range of the temperature of the steam.

From Par. 47 it is inferred that the phenomena occurring after cut-off are independent of the form of the cylinder and the relation between the volume and the clearance surface. The writer cannot believe that this is quite true. By changing the quality of steam, the volume, the clearance surface and the speed, there may be several engines having the same quality of steam at cut-off, but in which the rate of radiation of heat from cylinder walls into the steam will be quite different. Under the circumstances different values of n may be expected for the same quality of steam at cut-off. The mere fact that two or three engines of apparently dissimilar design give substantially similar values for n and X_c does not disprove the possibility of this variation, and until this point receives further attention, it cannot be taken for granted.

In Par. 64 it is stated that this method is the only one which measures the *rate* of steam consumption of an engine. A steam meter may be employed for the same purpose, and will probably be more accurate. In certain types of engines and with certain forms of indicators the error of the expansion line due to vibration will make the method very inaccurate. I do not believe that the method is sufficiently accurate to warrant its use in determining steam consumption, although it will probably be very useful in determining losses from leakage and in showing whether or not an engine is operating under the most economical conditions possible.

The method does not take account of leakage, and if there is any the steam consumption obtained will be lower than the true steam consumption by some unknown amount. Since there are forms of leakage which will not show when the logarithmic analysis is employed, it follows that there will be cases when the steam consumption obtained by this method will be too low and yet there will be no evidence of the fact. On this account, while the steam consumption may be determined with reasonable accuracy in many cases, one can never be sure whether the method is accurate or not, and it must therefore always be looked upon with suspicion.

In connection with Par. 74, how does this method check with itself when the steam consumption of the high-pressure and low-pressure cylinders of a compound engine are determined independently by its use? Does it give the same steam consumption for

both cylinders as it should? In the same paragraph the writer would call attention to the fact that the method cannot be applied when the load on the engine is varying, since the temperature conditions of the cylinder walls are those acquired from a previous revolution, and the quality at cut-off and the form of the expansion line will be different from what it would be were the engine operating with the same cut-off under steady load.

In Par. 84 is given a graphical method of determining the clear-

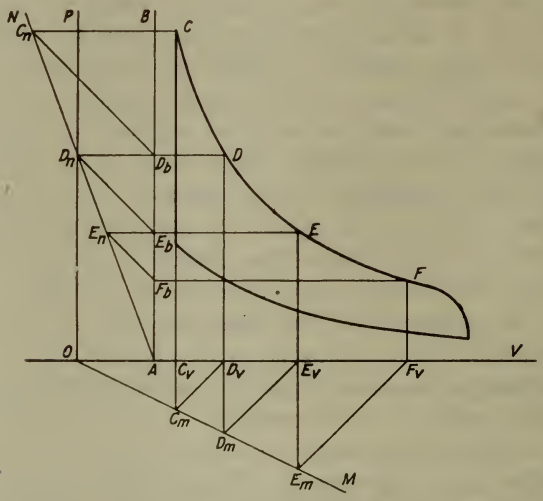


FIG. 32 GRAPHICAL METHOD FOR DETERMINING CLEARANCE

ance which is based on the assumption that the expansion line follows the law $PV^n = C$. The following is submitted as a preferable method:

In Fig. 32, CF is the expansion line of a gas engine card and OV the axis of zero absolute pressure. At any point A on this axis erect the perpendicular AB, and draw AN making any convenient angle to AB. Draw CC_n and draw C_nD_b making an angle of 45 deg. with AB. Through D_b draw D_nD and D_nE_b at 45 deg. Similarly, draw E_nE, E_nE_b, and F_bF. From F drop a perpendicular to OV and draw F_vE_n making an angle of 45 deg. with OV. From E_vD_m drop a perpendicular from D to determine D_m. Draw D_vC_m and drop a perpendicular from C to determine C_m. If the expansion line CF fulfils the equation $PV^n = C$, C_mD_m and E_m will lie on the same straight

line and that line will intersect OV at the point of zero volume. Erect *OP* at that point which is the origin of pressure and volume. This method [1] is more elegant, more accurate, and more simple than the method proposed in the paper.

There is no question but what the logarithmic analysis will show evidence of leakage in many cases. In certain cases, however, when there are simultaneous leaks of inlet and exhaust valves, it is quite possible that no leakage can be detected. This is also the case where the leakage takes place by the bodily transfer of water under the surface of a slide valve, which is the form of leakage most common in high-speed engines. There is no possibility of approximating the amount of leakage from the logarithmic diagram. Data in regard to the relation of the quality of the steam at cut-off and the exponent of expansion, obtained from engines equipped with valves known to be subject to leakage, should be regarded with suspicion.

It is a well-known fact that an engine which is apparently quite tight when standing, may have considerable leakage when it is running. So far as I can find from the paper, no attempt was made to determine the running leakage. It is unfortunate that Mr. Clayton did not check the quality of steam in the cylinder at cut-off by the thermoelectric methods employed by Callander and Nicholson. He did not state whether the engines tested have running leakage or not. It is desirable to have more evidence on this point, especially before using the method, for the purpose of determining the steam consumption.

If experience justifies the claims for this form of analysis, indicators which will draw the logarithmic diagrams directly upon the indicator card can be manufactured. The mathematical relations required are quite simple, and there ought to be no difficulty in constructing an accurate and simple apparatus for the purpose. Of course such an indicator will not draw correct diagrams unless the clearance of the engine has been determined and the apparatus adjusted accordingly. If, however, the indicator can be adjusted until the expansion lines are straight, it will be possible to use the apparatus without determining the clearance.

THE AUTHOR. It is gratifying to find that the development of the new methods of analyzing cylinder performance described in the paper and the conclusions reached regarding them should

[1] For the proof see Practical Thermodynamics, McGraw-Hill Book Co., pp. 33 to 35.

be so generally admitted and approved, especially since these methods involve a most radical departure from the beliefs and teachings which have been held for the last fifty or sixty years.

The only serious objection which has been brought out is the effect of valve leakage, i.e., the leakage of steam through the valve in single-valve engines which does not enter the cylinder, on the method of determining steam consumption.

The relative importance of this objection resolves itself into the actual amount that this form of leakage assumes in practice and the relative size and number of single-valve engines in service as compared with the total horsepower of steam engines used in plants where the highest economy is essential.

The relations of the values of n and the quality of steam at cut-off, given in Fig. 6, are primarily applicable to engines of the general type of that tested, that is, Corliss or four-valve engines in which all the steam passes through the cylinder.

The elimination of valve leakage was the motive which led to the use of a Corliss engine to determine the constants given, as explained in Par. 14. It is obvious that any steam lost by valve leakage has no effect upon the diagram and therefore cannot be accounted for. The importance of leakage of any kind in affecting the results of the steam consumption method is fully pointed out in Par. 67, 69, 73 and 74.

The steam consumption method is however a direct means of measuring valve leakage, since the steam and water present in the cylinder may be accurately accounted for. This method has since been used with a high degree of accuracy in connection with the measurement of boiler-feed to measure the valve leakage of a locomotive engine.

The single-valve engine is not an important factor in large steam engine plants where power must be produced at the lowest possible expenditure of steam. It is used more generally in installations where its low first cost is important and where its high steam consumption is not objectionable, as in small isolated plants having need of all exhaust steam for heating, in auxiliary machinery, and also in locomotives and marine engines because their service conditions demand the simplest possible type of engine. However the valves employed in locomotives and marine engines usually leak very little steam, measured as the proportion of the total amount used, on account of the high speed

of the engine giving large specific capacity for the size of the valve, and also because the large size of the valve enables a very close fit to be made.

The greatest field for the steam consumption method will be found in plants where the highest steam economy is essential, or where tests are impracticable due to service conditions, and in this field the small single-valve engine is relatively unimportant.

The engine tested by Mr. George Mitchell and quoted by Mr. Stevens was only 6 in. by 6 in. in size. The 80-h.p. engine tested by Mr. Stevens has a cylinder probably not larger than 12 in. by 12 in. in size. It is only in the very small single-valve engines, which are unimportant from a power standpoint, that valve leakage reaches the value of 20 per cent, a figure Mr. Stevens uses as an average for single-valve engines. Even small single-valve engines may have very little valve leakage as will be seen from the three examples of this type of engine given in Table 4. The two smaller engines, 14½ in. by 13 in. in size, were tested by Mr. George H. Barrus, and the test results together with the diagrams were taken from his book, Steam Engine Tests. The average difference of the steam consumption for these three engines, as determined from the diagram and the test consumption, is 7 per cent, all of them being within the 8 per cent given in the paper.

Mr. Stevens' estimate of the average amount of valve leakage shows that he has totally neglected to take into account the most important type of single-valve engine, the steam locomotive. It is absolutely impossible for the valves of locomotive engines to leak any considerable amount and still obtain the high steam economy which is found on test plants. When a locomotive valve does leak as much as 20 per cent a loud steady "blow" is heard between exhausts.

The effect of leakage upon the application of the steam consumption method may be summed up as follows: If a properly taken indicator diagram is obtained from a noncondensing nonjacketed Corliss or four-valve engine, and if the logarithmic diagram shows that there is no serious leakage, the constants of Fig. 6 will give the steam consumption as stated in Par. 74. If however the same procedure is followed for a single-valve engine of the same type, the result depends upon the amount of valve leakage, but usually with a fair sized engine and the

valves in good condition, the steam consumption may be obtained from the diagram to within a maximum difference of 8 per cent of the test results, as shown in Table 4.

The results obtained by Mr. Rice in his applications of the method corroborate the figures given in the paper.

The variation of the n-X_0 law with pressure mentioned by Mr. Moss has been verified further in work on locomotive engines. The distinction between different pressures rests upon the average coördinates of the points found at each pressure, although the average of all pressures gives results which are nearly as close.

Professor Heck's analysis of the heat transfer of series V of Table 1 corroborates the statement in the paper that n is a constant for ordinary conditions. The author has found, as Professor Heck mentions, that there is a tendency at very early cut-off for n to be high at first and then fall to a lower value. Curves No. 3 and 5 in Fig. 30 depart widely from the polytropic law due to the extremely low speed and probably also to steam-valve leakage during the latter part of expansion.

Mr. Marquis describes one of the broadest fields of usefulness for the new analysis, and shows that results may be obtained by its use which have not yet been obtained by means of present methods. A large number of locomotive tests have been analyzed since this paper was written.

Mr. Young questions the statement in Par. 8 concerning the usefulness of older methods of analysis. This statement is fully substantiated by the general results of the analysis. Previously no accurate method of obtaining the steam consumption from the diagram was in use, nor was any method known of determining leakage from the diagram with the engine in regular operation, and although several attempts had been made to determine leakage by the divergence of the expansion line from an equilateral hyperbola, yet the methods followed generally possessed no meaning and gave results of no value. The old method of obtaining the clearance of a steam diagram is fully explained in Par. 136, which shows its unreliability and that it is founded on wrong premises. No method of locating the cyclic events accurately was in use, especially for the events of a locomotive diagram.

The quality of the steam mixture at cut-off, called X_c, obtained

in the engine tested when using superheated steam, is below unity because the degree of superheat at the engine throttle was only from 50 deg. to 125 deg. fahr., and also because of the small 12 in. by 24 in. cylinder. Mr. Young probably has in mind locomotive cylinders of about 27 in. by 28 in., and degrees of superheat of from 200 to 250 deg. fahr. These conditions result in the steam being superheated about 50 to 90 deg. fahr. at cut-off.

The locomotive tests mentioned by Mr. Young have already been analyzed and the data will soon be available for use.

Mr. Stanwood questions the low values of X_0 obtained from the engine tested when saturated steam is used and compares them with the figures of Mr. Barrus for the same cut-off values of other engines. The engines mentioned by Mr. Barrus had cylinder displacements several times larger than the small engine tested, and the difference in the value of X_0 is due to small size which relatively increases the cylinder surface per cubic foot of displacement space. Mr. Stanwood is right in suspecting that some of these engines were tested by Mr. Barrus, in fact, about half of them were taken from his book, Steam Engine Tests.

The engine used was carefully tested for leakage by the usual standing tests and also by running the engine with steam on the crank end only, and taking off the cylinder head to observe piston leakage. It may be stated that valves and piston were practically tight when judged by these tests. The tightness of these parts was further demonstrated by the form of the logarithmic diagrams obtained from the engine, one of which is shown in Fig. 9.

Professor Ennis' high values of n for compression of steam are probably due to leakage from the steam chest into the cylinder at this point as he states in his discussion. Leakage through the steam valve however does not affect the early part of expansion, as stated by Professor Ennis, but only the latter part of expansion when there is considerable difference in pressure between the steam in the cylinder and that in the steam chest. This phase of leakage is treated in Par. 109, etc.

Professor Cardullo is correct in saying that the law $PV^n=C$ does not perfectly express the adiabatic expansion of steam. This law does express it however to within an average error of less than 1/5 of 1 per cent for saturated steam, as may be seen by referring to Bulletin No. 58, page 81, of the University of Illinois Engineering Experiment Station. However the question

as to whether the polytropic law $PV^n = C$ expresses the adiabatic expansion of steam has no direct bearing on the application of this law to the actual expansions occurring in practice.

The evidence gained from plotting over 1600 logarithmic diagrams from over 60 different engines using steam, air, gas and ammonia amply substantiates the statement that expansion and compression of elastic media obey the polytropic law except where certain exceptional conditions exist.

Professor Cardullo contends that since adiabatic expansion gives a straight line in the logarithmic diagram, it follows that the actual expansion cannot be straight. It must be remembered in this connection that theories are courses of thought which express the causes of facts, and that facts never conform to theories when these theories are incomplete and are based on wrong premises.

The present status of the actual forms of expansion curves in cylinders and the amount and direction of the heat transfer that has therefore taken place to give the curves is as follows: The facts gathered from 1600 examples prove that expansion and compression curves obey substantially the polytropic law under normal conditions of cylinder tightness, etc.; therefore it follows that any theories which lead to other conclusions are incomplete and are not supported by the facts, and hence these theories must be discarded or revised to express the facts.

The bend in the expansion line of Fig. 31 is the same as that caused by large leakage from the cylinder; tight engines do not give this abnormal effect.

Professor Cardullo's hypothesis concerning the relation between the volume and the clearance surface is fully answered by actual examples given in Table 4. Analysis Nos. 106*a* and 110 are two examples which prove that, given the same values of X_c, the values of n which result are practically the same regardless of cylinder size or form and of engine speed. In this matter, as in the matter of polytropic expansion, facts take precedence over theories.

When steam meters are used to measure the rate of steam flow to reciprocating engines, the makers of this apparatus state that the calibration obtained with steady steam flow no longer holds for the intermittent flow of steam engines, and that these meters have to be recalibrated by an actual test for each individual location to give reliable readings.

The steam consumption of the high and low-pressure cylinders of compound engines check each other very well in work done since this paper was written.

The varying temperatures in a cylinder due to swinging loads do not alter seriously the accuracy of the steam consumption method.

The clearance method offered by Professor Cardullo is the reverse of the old method of constructing graphically a curve of the form $PV^n=C$. It is shorter than the logarithmic method, but it is neither as accurate nor as simple. It is not as accurate for the reasons given in Par. 90, nor as simple because the trigonometric relations used involve a higher degree of mathematical education than the mere selecting of the straight line position on the logarithmic diagram according to directions.

With regard to a further amplification of the analysis, suggested by several of the members who have discussed the paper, the purpose in writing the paper was to establish the broad principles of the analysis, and to give sufficient examples of the application of these principles and some of the results attained by their application.

ADDENDUM

Subsequent to the preparation of the author's paper, a large number of tests upon locomotive engines were analyzed. The tests selected comprise those at the locomotive testing plant of Purdue University, Lafayette, Ind., on a 16-in. by 24-in. simple locomotive, conducted by Dr. W. F. M. Goss. The first series analyzed were made in 1904–1905 with saturated steam under different pressures [1] varying from 120 to 240 lb. gage; in the second series [2] a Cole superheater was used, giving about 160 deg. of superheat under various pressures. The analysis also included the St. Louis [3] tests run at the Louisiana Purchase Exposition in 1904 on a testing plant built by the Pennsylvania Railroad system, erected in the Transportation Building of the Exposition, and involving two simple and six compound locomotives; and also the Altoona tests, run on the same plant, removed to its permanent location in Altoona. Of the tests made here since 1904, two simple locomotives using saturated steam

[1] High Steam Pressures in Locomotive Service, W. F. M. Goss.
[2] Superheated Steam in Locomotive Service, W. F. M. Goss.
[3] Locomotive Tests and Exhibits, published by P. R.R. Co.

TABLE 7 GENERAL DATA FOR SIMPLE LOCOMOTIVES TESTED

Number or Name of Locomotive	Owner	Classification of Railroad Company	Type	Where Tested	Size of Cylinders, In.	Boiler Pressure	Quality of Steam Used	Type of Valves	Speeds Tested, R.P.M.	Number of Tests Analyzed
Purdue	Purdue University	Schenectady Nos. 2 and 3	American Eight Wheel	Purdue University, Lafayette, Ind.	16 x 24	120-160-180 200-220-240	Saturated and Superheated	Balanced Slide	97.5-146-195-243.5	104
7510	Penn. R.R.	K 2	Pacific	Altoona, Pa.	24 x 26	205	Saturated	Piston	80-100-120-160-180	28
3395	Penn. R.R.	K 29	Pacific	Altoona, Pa.	27 x 28	200	Superheated	Piston	100-120-160-180-240-280	18
5266	Penn. R.R.	E 2a	Atlantic	Altoona, Pa.	20½x 26	205	Saturated	Double-Ported Balanced Slide	80-120-160-200	13
1499	Penn. R.R.	H 6a	Consolidation	St. Louis, Mo.	22 x 28	205	Saturated	Balanced Slide	40-80-120-160	11
734	L. S. & M. S. R.R.	B1	Consolidation	St. Louis, Mo.	21 x 30	200	Saturated	Double-Ported Balanced Slide	40-80-160	15

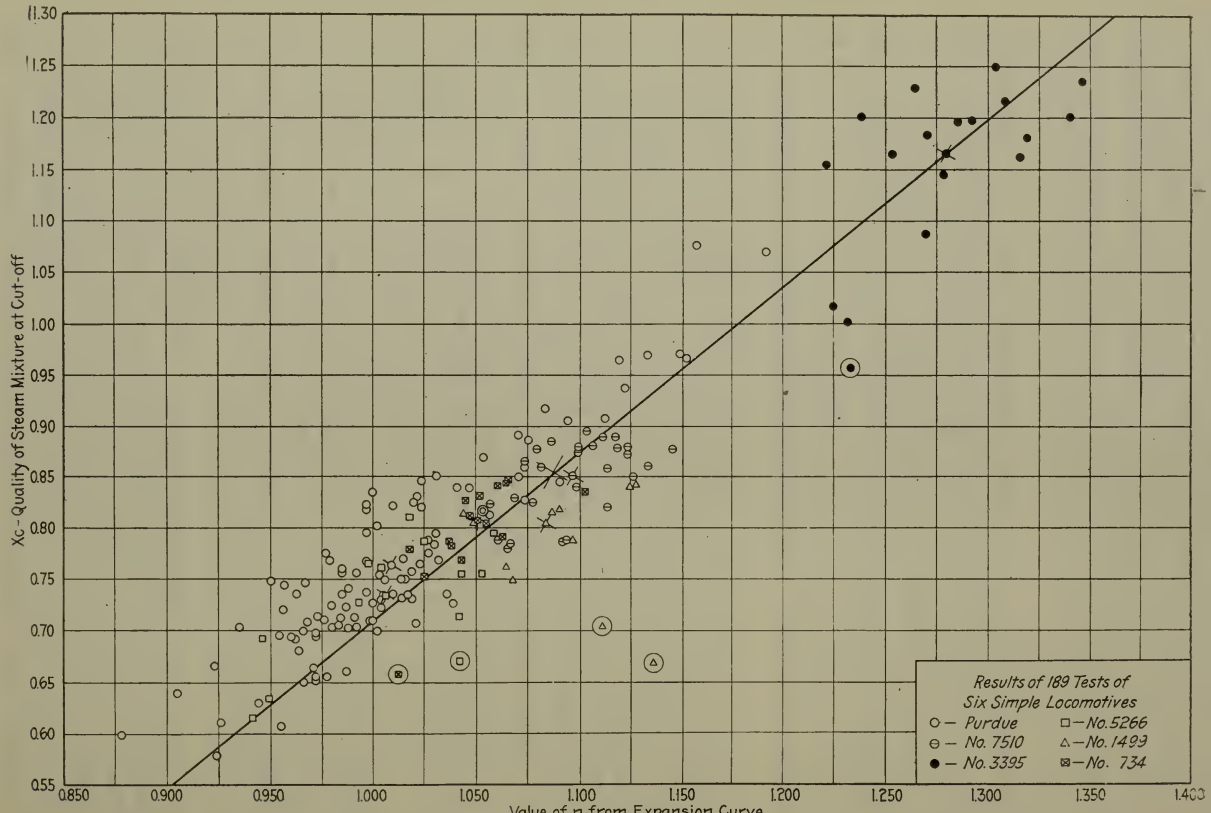


FIG. 33 RELATIONS OF X_c AND n FOR ALL SIMPLE LOCOMOTIVES TESTED

and one simple locomotive using steam superheated about 250 deg. Fahr. were analyzed.

The locomotives from which the tests were obtained comprise practically all important types in use in America and Europe, except the Mallet compound locomotive. Six simple and six compound locomotives are represented. A careful selection of the tests was made to insure reliability.

A total of 189 tests from six simple locomotives were analyzed, of which 104 were upon the Purdue locomotive. Almost every condition which is ever present in a simple locomotive cylinder is represented by these tests, except speeds below 40 and above 280 r.p.m. and lengths of cut-off over 50 per cent. The specifications of the locomotives and test data are given in Table 7.

Relations of X_c and n. All the values of X_c and n from the 189 tests of the six locomotives were plotted in Fig. 33, which proves the independence of the relations between these quantities from the effect of unduly varying pressures, speeds, sizes, types of valves and quality of steam used. The center of gravity, or average of the values of X_c and n was obtained for each locomotive by averaging the coördinates of X_c and n and plotting the result as a cross through the same kind of a point that was used to represent the tests of each locomotive. The center of gravity for all the locomotives was then determined in a similar manner from the centers for each locomotive and plotted as a large cross.

A straight line was then drawn through the center of gravity of the tests of all locomotives, and was made to pass through the center of the tests of locomotive No. 3395. Previous experience with Corliss engines, as recorded in the paper, indicated the use of a straight line for this purpose. The line as drawn represents the centers of the tests of all the locomotives with an average deviation of 2.4 per cent measured from the zero of X_c. The average deviation of the points above the curve is about the same as that of the points below the curve.

The center of gravity of all tests is shown by a concentric circle which is 2.4 per cent higher than the curve, due to the fact that the Purdue locomotive has 55 per cent of all the tests, and is the highest point in distance measured above the line. The points shown with a circle around them were not considered in drawing the line, as they were undoubtedly subject to certain inconsistencies in testing. The equation of the line selected is

$$X_c = 1.620\ n - 0.909$$

This equation expresses the results of all tests (except the five encircled points) to an average deviation of 5.0 per cent regardless of sign, and to a normal maximum deviation of about 9 per cent. It must be remembered, however, that all the errors of very short boiler tests are present, and their effect is included in the values of X_c. For this reason the differences given are believed to be considerably larger than actually existed.

Effect of Varying the Speed on the Relations of X_c and n. In order to examine the relations of X_c and n due to varying speed, the center

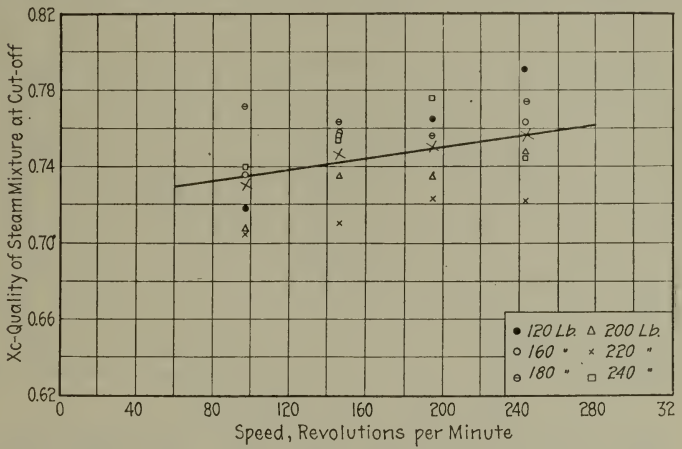


FIG. 34 RELATIONS OF X_c AND SPEED FROM THE PURDUE LOCOMOTIVE FOR THE CONSTANT VALUE OF $n=1.000$

of gravity of each group of tests run at one speed was obtained and a line drawn through this center parallel to the full line (Fig. 33) representing the average relation for all the locomotives. The point of intersection of this parallel line with the ordinate for $n=1.000$ thus gives the value of X_c for the constant value of $n=1.000$ at the speed examined.

Points of intersection for the groups of tests upon the Purdue Locomotive are plotted in Fig. 34 and for all the simple locomotives in Fig. 35. These show that the average effect of increasing the speed was to increase the value of X_c for constant values of n. In Fig. 35 the points were divided into two groups and the center of gravity of each group determined as indicated by the crosses. The line was then drawn through the centers. In Fig. 35 average deviation of the points irrespective of sign is 3.2 per cent (measured from

zero of X_c), each point being weighted in proportion to the number of tests it represented, except the Purdue tests, which were rated at four tests each.

From Fig. 33 it is seen that the line crosses the value $n=1.000$ at $X_c=0.711$. On drawing the line $X_c=0.711$ in Fig. 35 it is found to intersect the line representing the relations of X_c and speed at 143 r.p.m. This intersection indicates that the average relation of X_c and n represented by the line $X_c=1.620\ n-0.909$ is true for a speed of betwwen 140 and 150 r.p.m. It also indicates: (*a*) that the average points obtained at speeds below 140 r.p.m. fall below the

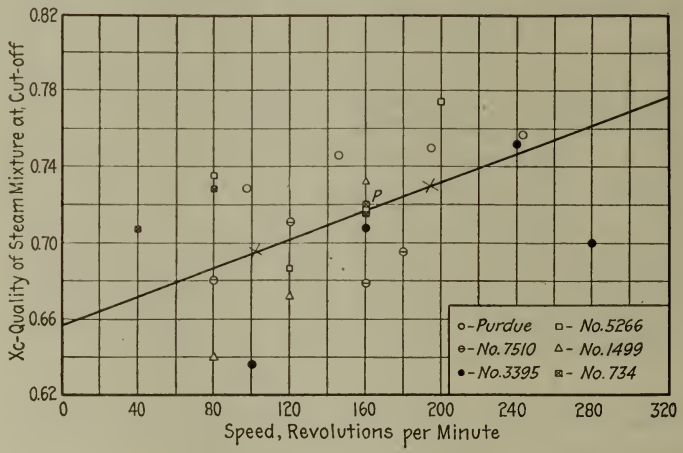


FIG. 35 RELATIONS OF X_c AND SPEED FROM ALL SIMPLE LOCOMOTIVES FOR THE CONSTANT VALUE OF $n=1.000$

line; (*b*) that the average points obtained at speeds above 150 r.p.m. fall above the line; (*c*) that by making corrections for the speed at which the indicator diagrams were obtained, the average difference resulting from determining the value of X_c by the equations given is reduced from 5 per cent to 3.2 per cent.

The equation $X_c=1.620\ n-0.909$ was modified therefore by adding a corrective term for speed which expresses the relation of X_c and speed shown in Fig. 34. The modified equation is

$$X_c=1.620\ n-0.909-0.00037\ (143-S)$$

where S = speed in r.p.m.

The points for the other speeds were plotted in like manner and divided into four groups of six points each and the center of gravity

of each group found as indicated. The centers were joined by a smooth curve. The curve as drawn indicated that the value of X_c at constant values of S and n reached an average maximum value for a cut-off pressure of about 105 lb. absolute, and that above 125 lb. a straight line was obtained.

It appears therefore that as the cut-off pressure increases the value of X_c decreases for constant values of S and n.

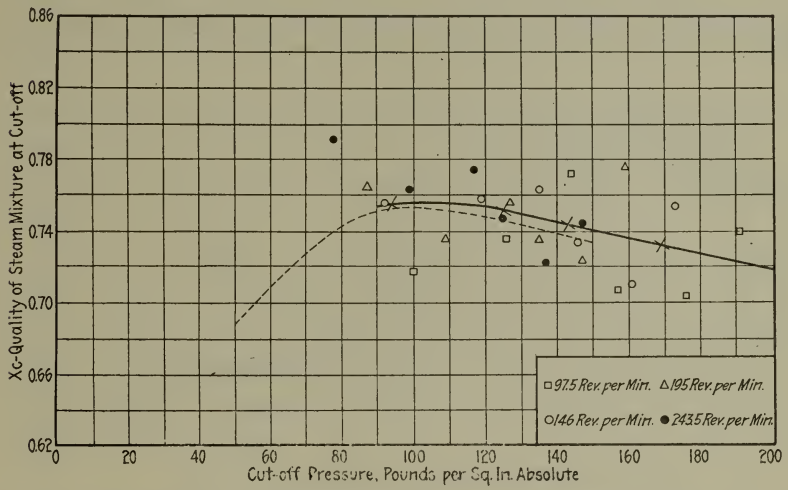


FIG. 36 RELATIONS OF X_c AND CUT-OFF PRESSURE FROM THE PURDUE LOCOMOTIVE FOR CONSTANT SPEEDS AND THE VALUE OF $n = 1.000$

The effect due to change of cut-off pressure on the relation of X_c and n was compared with the same effect found in the previous tests on a Corliss engine and the relation for this engine for the value of $n = 1.052$ and the speed of 120 r.p.m. was drawn in dotted as shown.

The result of this comparison is a striking corroboration of the curve as drawn. The curves are almost exactly parallel in the range common to both. Both have a maximum point only 5 lb. apart, and both have about the same slope for pressures above 120 lb. The form of the curve representing the effect of cut-off pressure is thus fairly well established.

The points in Fig. 35 were examined for the effect of cut-off pressure in a similar way to that described for the Purdue tests, and the results are given in Fig. 37. The centers of gravity obtained in Fig. 36 were transferred to Fig. 37 to represent the Purdue tests. The

points of Fig. 37 give the average relation for all locomotives of X_c and cut-off pressure for a series of constant speeds and for $n=1.000$. The group of points obtained show the same trend as the Purdue tests, although the vertical displacement due to speed is present and makes the relation somewhat indefinite.

It was assumed that the Purdue tests shown in Fig. 36, because of their large range of cut-off pressures and similarity to previous relations, represented the form of the curve, but was placed too high on the plot to represent the average relation for all locomotives. Therefore the center of gravity of all the points of Fig. 37 was found

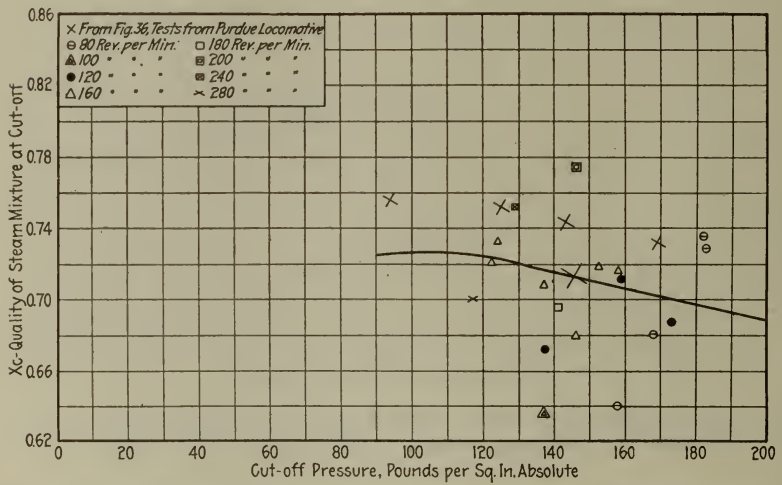


FIG. 37 RELATIONS OF X_c AND CUT-OFF PRESSURE FROM ALL SIMPLE LOCOMOTIVES FOR CONSTANT SPEEDS AND THE VALUE OF $n=1.000$

and a curve drawn through this center similar in form to the Purdue relation.

It is seen from Fig. 37 that the range of cut-off pressures in simple locomotives is between 120 and 180 lb. absolute, while the majority lie between 130 to 160 lb. The average cut-off is about 145 lb., this point being close to the average value of $X_c=0.711$ for $n=1.000$ given in Fig. 33. It is also evident that the difference in the value of X_c for the average cut-off, 145 lb., to an average high value, 165 lb., is only 1.6 per cent while to an average low value, 125 lb., it is only 1.3 per cent.

The effect of varying the cut-off pressure on the relation of X_c and

n is so small of itself, and so much smaller than the effect due to change of speed, that it need not be considered.

Effect of the Use of Saturated or Superheated Steam. The use of superheated steam in cylinders in place of saturated steam results in a higher value of X_c for the same conditions of speed and cut-off; and this in turn results in a higher value of n. When the same value of X_c is obtained at the same speed with either saturated or superheated steam, the same value of n has been found to result.

Effect of Varying the Length of Cut-off. The length of cut-off used in the tests varied from 12 per cent to 50 per cent. It has been found that, other conditions being the same, if a given value of X_c occurs in one test at 12 per cent cut-off, and in another test on the same locomotive at 46 per cent cut-off, the value of n resulting is the same, showing that the length of cut-off has no measurable effect on the relation of X_c and n.

Effect of Varying the Back Pressure. The back pressure of the steam in the cylinders varied from 15 to 36 lb. absolute, a range of 21 lb. Varying the back pressure between these limits appears to have no noticeable effect on the relation of X_c and n. When the back pressure is as high as 100 lb. absolute, however, as in the high pressure cylinders of compound engines, the relation of X_c and n is changed considerably.

INVESTIGATION OF COMPOUND LOCOMOTIVES

In the investigation of compound locomotives a selection was made of 43 tests of high-pressure cylinders and 34 tests of low-pressure cylinders. The data for the locomotives tested are given in Table 8. A new variable entered in these tests not present in the tests of the simple locomotives, viz., widely varying back pressures due to variable receiver pressures. The methods employed in the analysis of the diagrams were substantially the same as already recorded, certain few modifications being necessary which will not be explained in this brief synopsis.

Relations of X_c and n for High-Pressure Cylinders. The values obtained exhibit a much larger variation in position than from the simple locomotives in Fig. 34, due to the variable receiver pressure. The slope of the line, also, could not be definitely determined, but in the light of previous experience it is believed that it should be similar to that obtained from simple locomotives. A line was therefore drawn as in Fig. 38, the equation of which is:

$$X_c = 1.620\ n - 0.827$$

TABLE 8 GENERAL DATA FOR COMPOUND LOCOMOTIVES TESTED

Number	Owner	Classification of Railroad	Type	Place Tested	Size of Cylinders, Inches	Boiler Pressure	Quality of Steam Used	Type of Valves	Speeds Tested	Number of Tests Analyzed	
										H.P.	L.P.
585	M.C.R.R.	W	Consolidation	St. Louis, Mo.	2 Cylinders 23 & 35x32	210	Saturated	H.P. Piston; L.P. Double-Ported, Balanced Slide Piston	40-80-160	9	9
929	A. T. & S. F. R.R.	900	Santa Fé	St. Louis, Mo.	4 Cylinders 19 & 32x32	225	Saturated		40-60-80	9	9
2512	Penn. R.R.	De Glehn Compound	Atlantic	St. Louis, Mo.	4 Cylinders $14\frac{3}{16}$ & $23\frac{11}{16}$x$25\frac{1}{4}$	225	Saturated	H.P. Balanced D Slide. L.P. Not Balanced Piston	80-160-240-280	8	5
535	A. T. & S. F. R.R.	507	Atlantic	St. Louis, Mo.	4 Cylinders 15 & 25x26	220	Saturated		80-160-240-280	8	10
628	Royal Prussian Railway	58	Atlantic	St. Louis, Mo.	4 Cylinders $14\frac{5}{12}$ & 22x$23\frac{5}{8}$	200	Superheated	H.P. Piston; L.P. Double-Ported, Balanced Slide Piston	80-160-240-280	9	9
3000	N.Y.C. & H.R. R.R.	11	Atlantic	St. Louis, Mo.	4 Cylinders $15\frac{1}{2}$ & 26x26	220	Saturated		80-160-240-280-320	9	11

The average deviation for all points from this line is 8.0 per cent, while the normal maximum deviation is 12.0 per cent.

The effect of change of speed is shown in Fig. 39, the modification of the equation for the relations of x_c and n being as follows:

$$X_c = 1.620\ n - 0.827 - 0.00034\ (150 - S)$$

In Fig. 40 are plotted the lines for change of pressures at constant speeds. It was found that raising the back pressure, and lowering

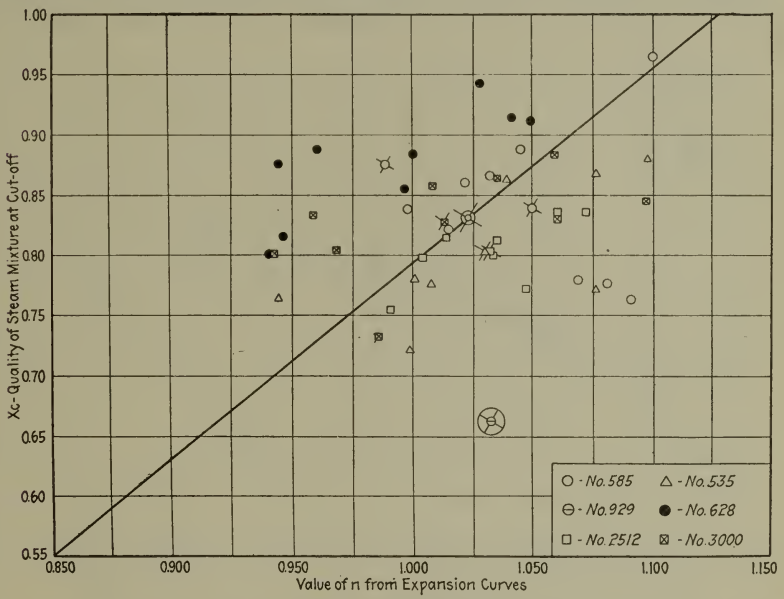


FIG. 38 RELATIONS OF X_c AND n FOR HIGH-PRESSURE CYLINDERS OF COMPOUND LOCOMOTIVES

the cut-off pressure both increased the value of X_c for the same value of n. In plotting Fig. 40 corrections for speed were made, based on Fig. 9. The equation for the line for the high-pressure cylinder in Fig. 40 is

$$X_c = 1.620\ n - 0.827 - 0.00034\ (150 - S) + 0.0031\ (102.5 - R)$$

where R is the range of pressure from cut-off to the back pressure in lb. per sq. in.

The average value of X_c is obtained at 102.5 lb. range of pressure; the value of X_c is higher for ranges of pressure below 102.5 lb. and lower for ranges above 102.5 lb. The average deviation of the points from the line is 6.2 per cent.

Relations of X_c and n for Low Pressure Cylinders. The values of X_c and n obtained are all plotted in Fig. 41. The line shown in Fig. 41 was drawn as outlined for the high pressure cylinders, using the same slope as the line obtained for the simple cylinders. The equation of the line is

$$X_c = 1.620\ n - 0.833$$

The average deviation of all the points from this line is 7.8 per cent.

Effect of Varying the Speed. The points obtained which show the effect on the relation of X_c and n are plotted in

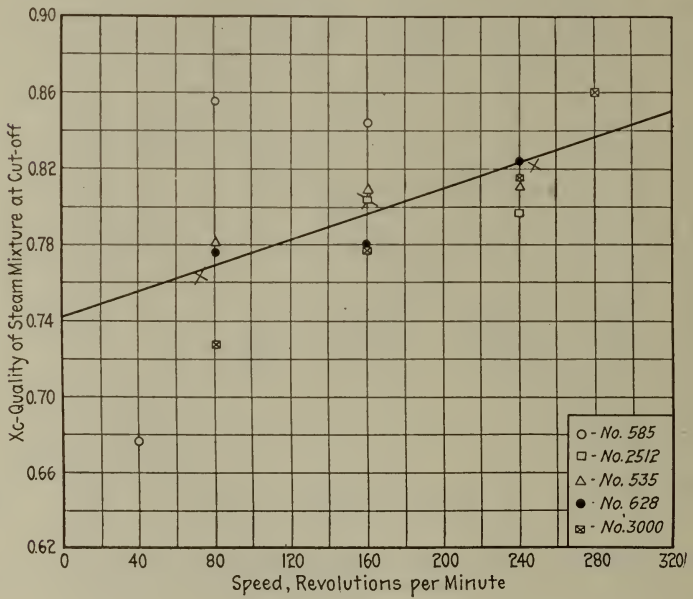


FIG. 39 RELATIONS OF X_c AND SPEED FOR HIGH-PRESSURE CYLINDERS OF COMPOUND LOCOMOTIVES FOR CONSTANT VALUE OF $n = 1.000$, AND FOR CENTER OF TESTS CORRECTED TO AVERAGE LINE

Fig. 42. The equation after correcting for the effect of change of speed is

$$X_c = 1.620\ n - 0.833 - 0.00069\ (154 - S)$$

The average deviation of the points from this line is 4.5 per cent.

Effect of Varying the Range of Pressure. The points obtained which show the effect on the relation of X_c and n of changing the range of pressure are plotted in Fig. 40. The equation as further modified by this effect is

$$X_c = 1.620\ n - 0.833 - 0.00069\ (154 - S) + 0.0047\ (30 - R)$$

The average deviation of the points from this line is 3.3 per cent. This extremely close agreement shows that the effects of varying the speed and range of pressure comprise all of the important variables.

DISCUSSION OF RESULTS

Effect of Varying the Range of Pressure on the Relations of X_c and n. The effect of lowering the range of pressure from cut-off to the back

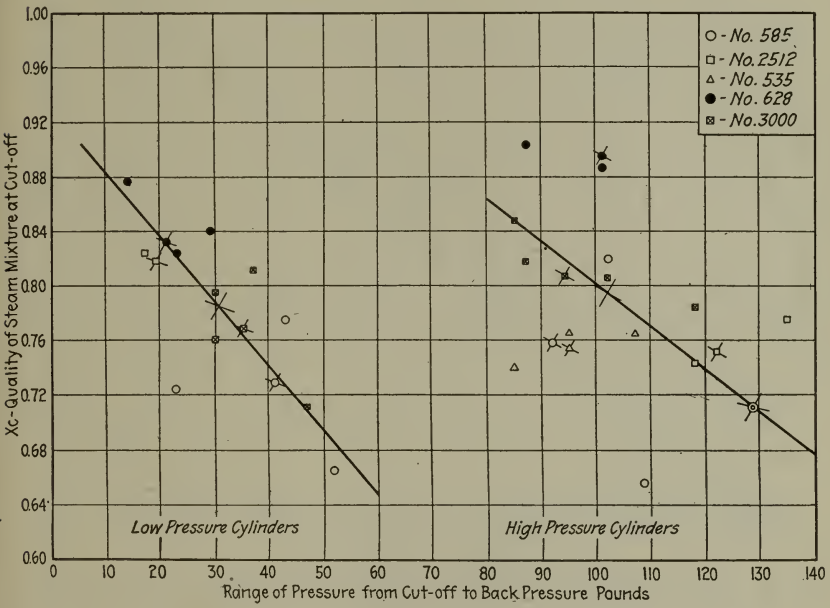


FIG. 40 RELATIONS OF X_c AND RANGE OF PRESSURE (CUT-OFF TO AVERAGE BACK PRESSURE) FROM COMPOUND LOCOMOTIVES FOR CONSTANT SPEEDS AND THE VALUE OF $n = 1.000$

pressure is always to give smaller values of n for the same value of X_c, other conditions remaining the same. The effect for simple locomotives is small, as will be seen in Figs. 36 and 37; it is larger for the high pressure cylinders of compound locomotives, as is shown in Fig. 40; it is still larger for low pressure cylinders, as is also shown in Fig. 40. The slopes of the lines in these figures are 0.12, 0.76, and 1.17, respectively, showing the increasing effect of this variable as the range of pressure decreases.

Effect of Varying the Speed on the $n-X_c$ Relations. The effect on the relations of X_c and n of increasing the speed is to give higher values of X_c for the same value of n, other conditions remaining the same.

Valve Leakage. The only phenomenon which accounts consistently for these results is the leakage of steam directly through the valves without having entered the cylinder. Experiments show that valve

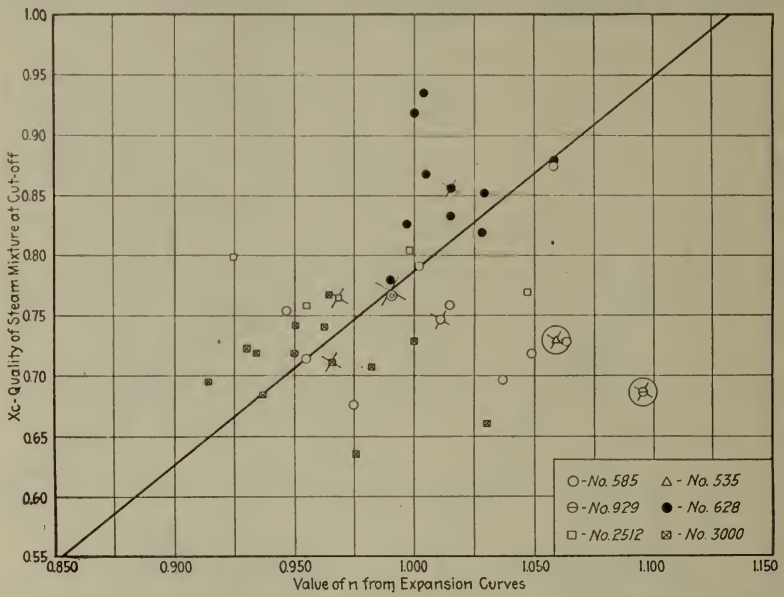


FIG. 41 RELATIONS OF X_c AND n FOR LOW-PRESSURE CYLINDERS OF COMPOUND LOCOMOTIVES

leakage in a given engine is constant per unit of time and therefore that it diminishes relatively with increase of engine speed.

Relatively speaking, simple locomotive engines may be classed as constant pressure and variable speed, while compound locomotive engines may be classed as variable pressure and variable speed. In the former, therefore, the range of speed has an important bearing on the $n-X_c$ relation, while pressure may be neglected; whereas in the latter both range of pressure and speed are important.

That is, it is probable that there is very little if any effect on the $n-X_c$ relations of varying the speed, but the existence of leakage and its increasing relative amount when speeds are decreased lead

to an apparent value of X_c which is lower than the real value for the steam actually admitted to the cylinders. It is also probable that the value of X_c for engines having the larger values of valve leakage would increase faster with increase of speed than engines having very small leakage, because the increase of speed would increase the apparent value of X_c at a much faster rate. This condition is exactly the one which occurs, as is shown in Figs. 33, 34, 35, 39 and 42.

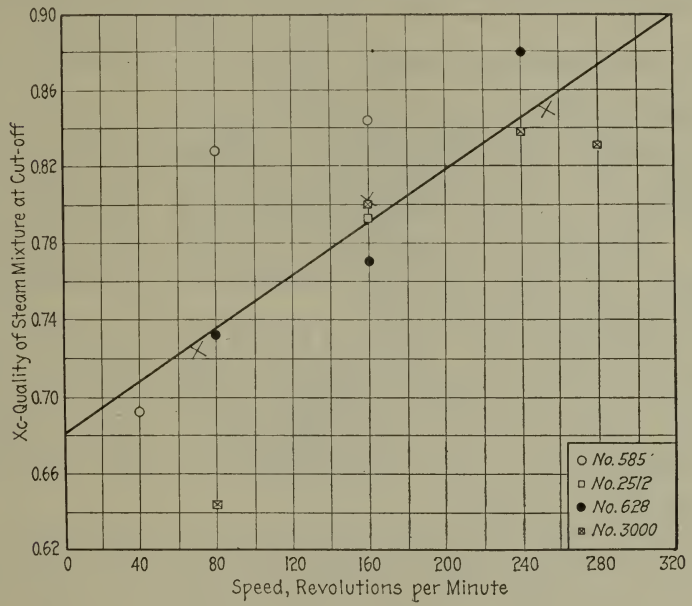


FIG. 42 RELATIONS OF X_c AND SPEED FOR LOW-PRESSURE CYLINDERS OF COMPOUND LOCOMOTIVES FOR CONSTANT VALUE OF $n=1.000$, AND FOR CENTER OF TESTS CORRECTED TO AVERAGE LINE

The slope of the line representing the effect of varying the speed for any one locomotive is apparently the best index which exists as to the amount of valve leakage.

A striking case of the effects of valve leakage is to be found in comparing the performance of locomotives Nos. 929 and 585, which had cylinders of almost exactly the same size and were tested under practically the same conditions.

The average steam consumption of No. 929, however, was 23.8 lb. per i.h.p., while that of No. 585 for the tests run at the same

speeds was 19.9 lb., showing that No. 929 consumed 19.6 per cent more steam than did the No. 585 for the same conditions.

The average quality of X_c in the high pressure cylinders of the No. 929 was found to be 0.836, as computed from the values of n, S, and R. In a similar manner the value of X_c for the low pressure cylinders was found to be 0.817, making the average value of X_c for both the high and low pressure cylinders 0.827.

The average apparent value of X_c from the tests for both high and low pressure cylinders was 0.674. Since this value of X_c is lower than 0.827, more steam to the amount $\frac{0.827-0.674}{0.827}$, or 18.5 per cent, was used than was accounted for in the cylinders.

The results of these computations is a striking proof that not only was large valve leakage taking place, but that its amount has been established by two independent methods which corroborate each other. Subsequent examination of the valves showed that the leakage as determined was to be accounted for by the design and condition of the valves and ports.

The analysis for the low points in Figs. 8 and 11 shows that the variations in the vertical distance of the center of tests for each locomotive from the average line is almost certainly due to the fact that the valves of each locomotive have their own rate of leakage, the leakage for some valves being less than the average, while that of other valves is greater.

Referring to Fig. 33, the relatively high point of the Purdue locomotive and its flat speed effect curve in Fig. 34 probably means that its valves were tighter than any of the others due to its being kept in laboratory condition. On the other hand the low point of No. 1499 in Fig. 33 and its steep speed effect curve in Fig. 35 is probably due to the fact that this locomotive was new and had not been broken in before it was tested, consequently the valves had not worn to a good surface, and the average leakage was larger than that for any of the other simple locomotives tested.

If it is desired to test an engine for valve leakage, the locomotive should be tested on a test plant or under road conditions where the steam used for other purposes is known. The steam used by the engines is accounted for by the $n-X_c$ method, the total water fed to the boiler is measured, and this includes the amount used in the cylinders; the amount that leaked through the valves over the average leakage of the valves when in good condition is accounted

for by the difference between the two measurements if the boiler is tight.

DEGREE OF ACCURACY

The steam consumption of all of the 185 tests of the six simple locomotives examined may be recomputed from the relations of X_c and n to an average difference of 3.7 per cent from the test results.

The average difference in the tests of the compound locomotives for the high pressure tests is about 8 per cent and for the low pressure tests it is 3.8 per cent.

It is extremely probable that at least one fourth of the per cent difference given is due to irregularities in measuring the difference in the height of the water in the boiler gage glass at the start and finish of the test. It must be remembered that all of the boiler tests were only $\frac{1}{2}$ to 3 hours in length, so that an error in the gage glass correction means a noticeable difference in the actual water evaporated. If one fourth of the difference is credited to errors of the boiler measurements then the average differences are 2.8 per cent, 6 per cent and 2.9 per cent for the simple cylinders, the high-pressure and low-pressure cylinders respectively. These differences show that the $n-X_c$ method is accurate enough for all the purposes of locomotive tests. The high difference of the high pressure cylinders is due to difficulty of obtaining good values of n because of inertia in the expansion curves, and also to variable valve leakage.

Steam for Heating and Miscellaneous Purposes. Locomotives in winter passenger service use such a large proportion of the steam generated by the boiler in heating the train, running the air pump, etc., that in combination with the increased train resistance in cold weather, schedules are only maintained with great difficulty. What proportion of the steam generated by the boiler is used for these purposes? It is practically impossible to measure the steam used in this way by any methods heretofore employed.

The $n-X_c$ method may be employed for this purpose by accounting for the difference between the total water fed to the boiler and that used by the engines. The difference was used to heat the train, to operate the air-pump, head-light sets, train-lighting sets, blower, and it also includes the steam and water lost through the safety valves, blow-off valves, and boiler leaks. The $n-X$ method provides the only means believed to exist by which these uses of steam for auxiliaries or that lost may be satisfactorily measured.

No. 1348

THE DESIGN AND MECHANICAL FEATURES OF THE CALIFORNIA GOLD DREDGE

By Robert E. Cranston, Sacramento, Cal.
Member of the Society

This paper deals only with that branch of gold dredging which comes directly within the province of mechanical engineering, and the designs described are those commonly used in California. No attempt is made to cover the whole field, and the author has recorded only his personal observations of the more important mechanical features.

2 In designing a gold dredge the first essentials to be determined are the quantity of material to be handled and the depth to which it will be necessary to dig. To a lesser degree, the size and quantity of boulders, quantity of clay, nature and contour of bed rock, contour of the surface, whether the dredge is to be operated in a river or in an inland pond, how high above the water line it will be necessary to maintain the bank, the nature and consistency of the gravel, distance from a railroad or other cheap transportation, costs of lumber, labor, electric power, supplies, etc., condition of wagon roads, distance and transportation facilities to a first class machine shop, climate, distance to nearest public power line, and possibilities of developing electric power on or near the property, are important. If electric power is not available, then the costs of the various fuels for generating steam or running gas engines must be considered, and if the dredge is to be operated inland, how much water is available, and any other data peculiar to the property.

3 All the above information is requisite for the proper design of a dredge to do the most efficient work. A designer should insist on a full knowledge of all the conditions, otherwise his product is likely to be more or less unsuited to the work it has to do.

4 Beginning with the capacity required, the dimensions of the bucket bases, pins, lips, etc., and the horsepower of the digging

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motor are determined. The depth of ground determines the length of ladder, which is modified by the height to which it is desired to elevate in order to wash and dispose of the gravel properly. The screen, pumps and stacker are designed according to the quantity and quality of coarse material and the height to which it is to be stacked. The other machinery is designed to suit the local conditions, and the hull is planned to support the machinery properly, its dimensions varying greatly, from the small capacity shallow diggers to the 16-ft. dredges excavating 60 or 70 ft. below water line.

5 The mechanical equipment of the California gold dredge as it is today has in it much to be desired in the way of correct dimensions and the relative strengths of its component parts. Many of these parts should be made stronger, likewise there is much waste material which could just as well be done away with. Notwithstanding these admitted faults the California gold dredge has during the last few years shown vast improvement over the earlier attempts, and with all its frailties has many good points in its make-up which might be profitably studied by engineers called upon to design similar pieces of machinery.

6 The first successful gold dredges in California were patterned after the New Zealand type. A brief description of one of these, built in 1897 and 1898, will not perhaps be out of place (Fig. 1).

7 The hull was built of wood of a rectangular cross-section, having an opening or well extending along the center line part way back from the bow. Near the after end of this well was built an elevated framed structure or main gantry which served to support one end of the digging ladder, the other end being suspended by wire rope blocks from a head frame or "bow gantry" spanning the well. This ladder was a plate girder upon which were mounted rollers to carry the bucket line. The bucket line was composed of alternate links and buckets pinned together to form an endless chain. The buckets passed over a four-sided flanged tumbler keyed to a shaft and driven by a spur gear mounted on the main gantry. At the lower end of the ladder another tumbler served as an idler. In going over the upper tumbler the buckets dumped into a hopper leading into an inclined revolving screen with $\frac{3}{8}$-in. to $\frac{5}{8}$-in. perforations. A spray pipe ran lengthways through the screen, throwing jets of water upon the gravel and washing the finer particles and gold through the screen holes upon gold-saving tables. These tables were covered with cocoanut matting held in place by expanded metal and were arranged so as first to distribute the fine material over a considerable area and

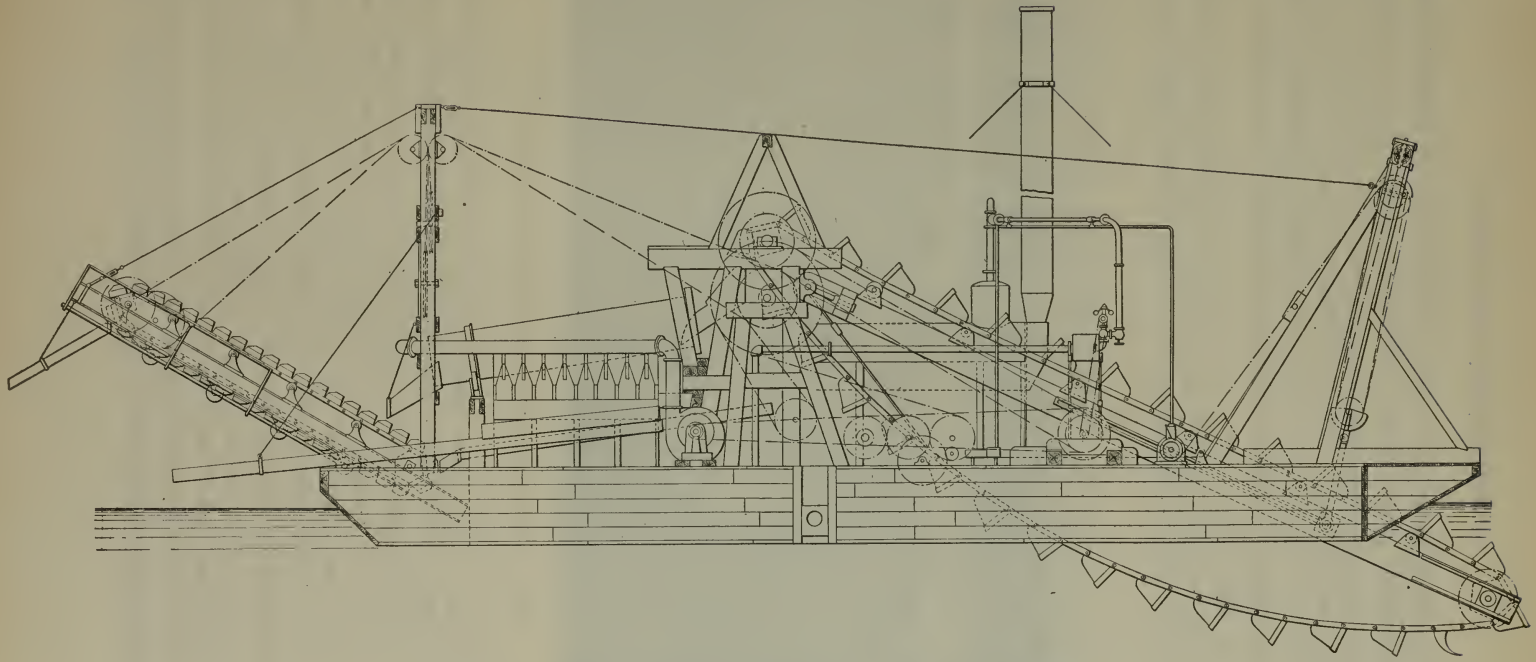

Fig. 1 First Successful California Dredge, 1897–1898

then discharge it at the stern; the oversize went to a bucket elevator and was stacked some distance aft of the dredge. The dredge was held up to its work by a head line which could be reeled in by means of a drum on the dredge. Side movement was secured by lines, two at the bow and two at the stern, each running to a separate drum. The motive power for these dredges was steam, which was soon replaced by electricity.

Three years before this type of dredge was introduced in California, one of the type known as the double-lift was built near Ban-

FIG. 2 SINGLE-LIFT CALIFORNIA GOLD DREDGE, 1900

nock, Mont. The main gantry was lower than in the New Zealand type. A revolving screen was used, but the holes were much larger, so that rocks 4 in. to 6 in. in diameter would pass through. The oversize was dumped at the side of the dredge, and the undersize allowed to collect in a sump from whence it was elevated by means of a centrifugal pump into a sluice running aft from near the throat of the well. At the stern was a pivot connection with another sluice mounted on an auxiliary scow. This scow could be moved from side to side, one end of its sluice being pivoted at the stern of the dredge and the other moving in the arc of a circle, thus distributing the gravel over a considerable area. Instead of a head line, spuds

were used to hold the dredge up to its work, the end of the digging ladder moving in the arc of a circle with the spud as its center.

9 About this time several dipper dredges were built, but although some were more or less successful they did not prove to be as efficient as the endless chain type and were soon abandoned.

10 In 1899 a double-lift dredge was built at Oroville, and soon after a similar dredge was built on the American River, the latter, however, being of the single-lift type (Fig. 2), using close-connected buckets (every link in the chain being a bucket instead of links and buckets alternating) with spuds, no stacker, but having its main

Fig. 3 First California Type Gold Dredge, 1901

gantry high, the buckets dumping into a revolving screen with large holes. This screen was raised high enough so that the finer material fell directly into a fore-and-aft sluice; the oversize dumped into stone shoots discharging on both sides of the dredge.

11 A judicious combination of the best features of the New Zealand and the double and single-lift types described above, together with several new ideas not used in any of the earlier dredges, resulted in a dredge being built some time in 1901, near Oroville, Cal. (Fig. 3), equipped with a shaking instead of a revolving screen, the holes in which were small, the fine material going to gold-saving tables as in the New Zealand type, but with Hungarian riffles replac-

ing the cocoanut matting and expanded metal. The oversize was stacked at the stern by means of a belt conveyor instead of a bucket

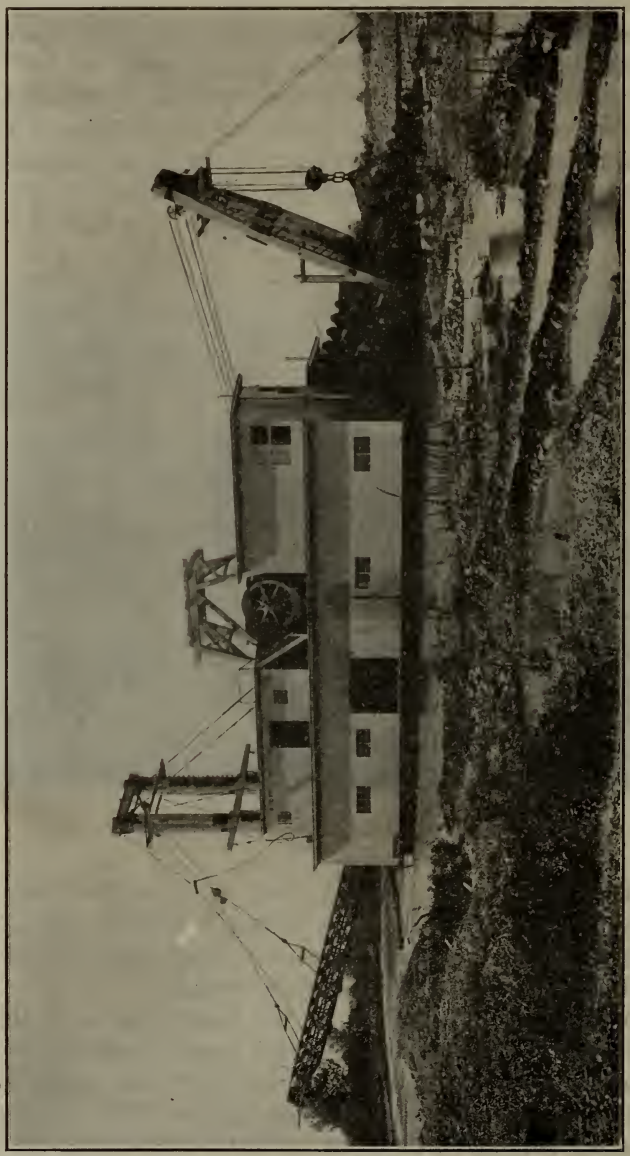

FIG. 6 MODERN 5-CU. FT. GOLD DREDGE, 1907

elevator. Spuds and close-connected buckets were used as in the single-lift type. This general arrangement became known as the

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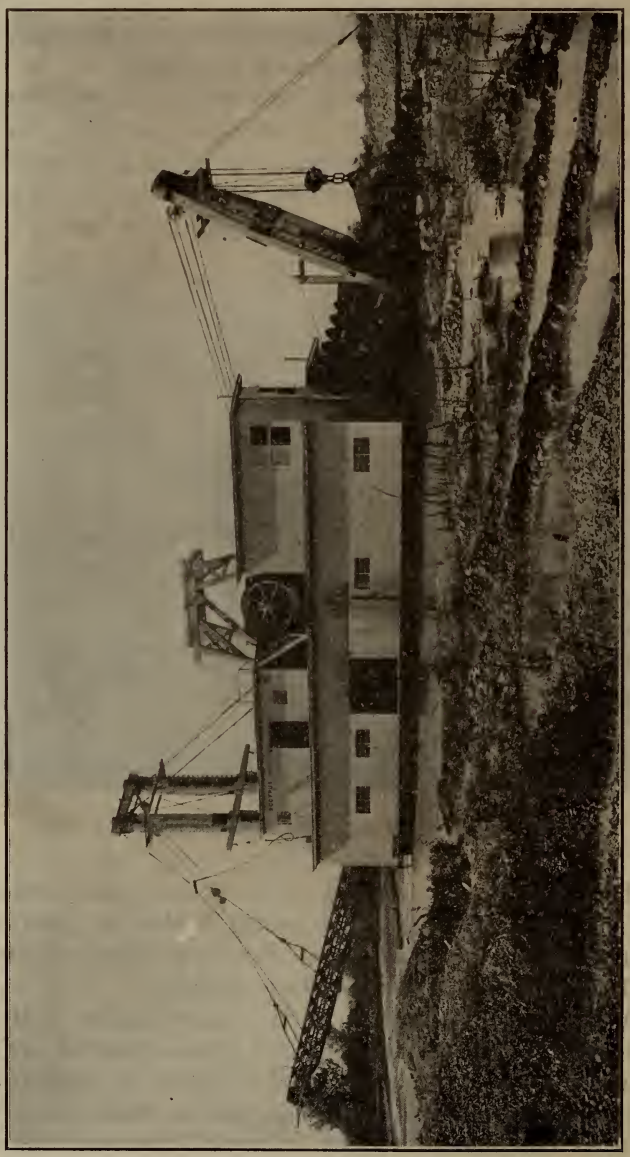

Fig. 6 Modern 5-Cu. Ft. Gold Dredge, 1907

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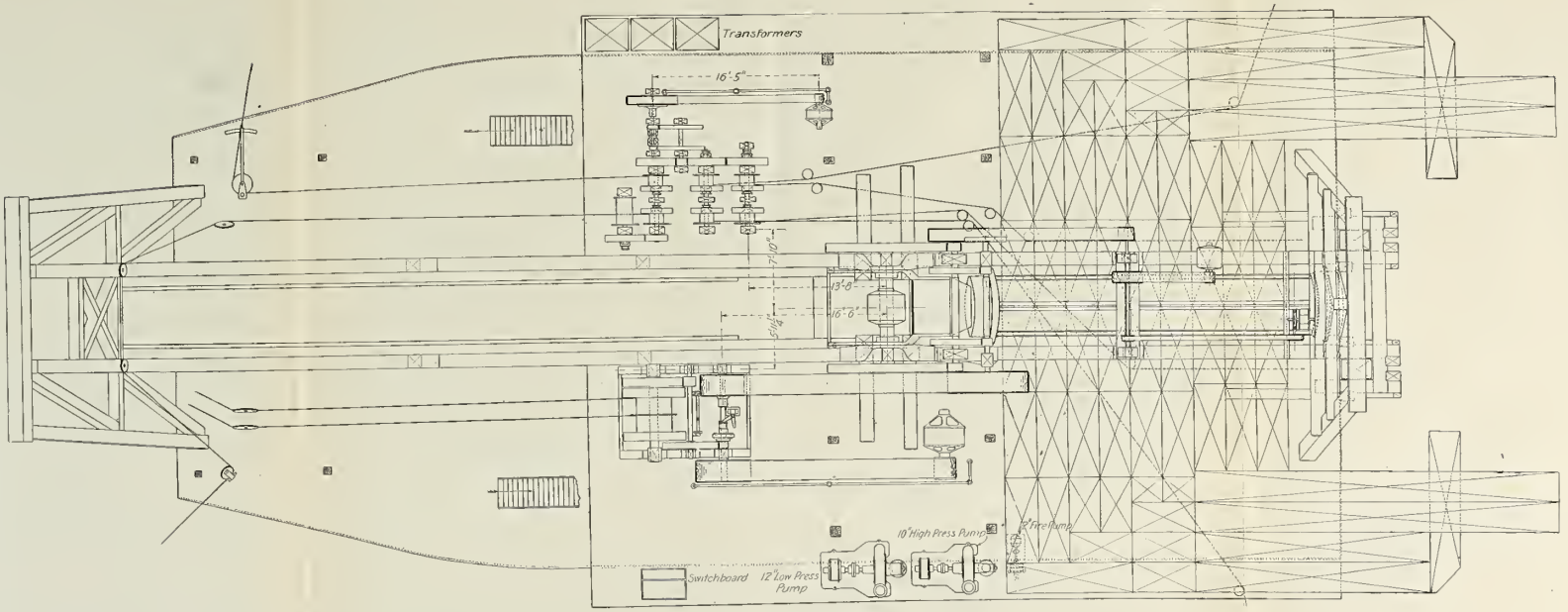
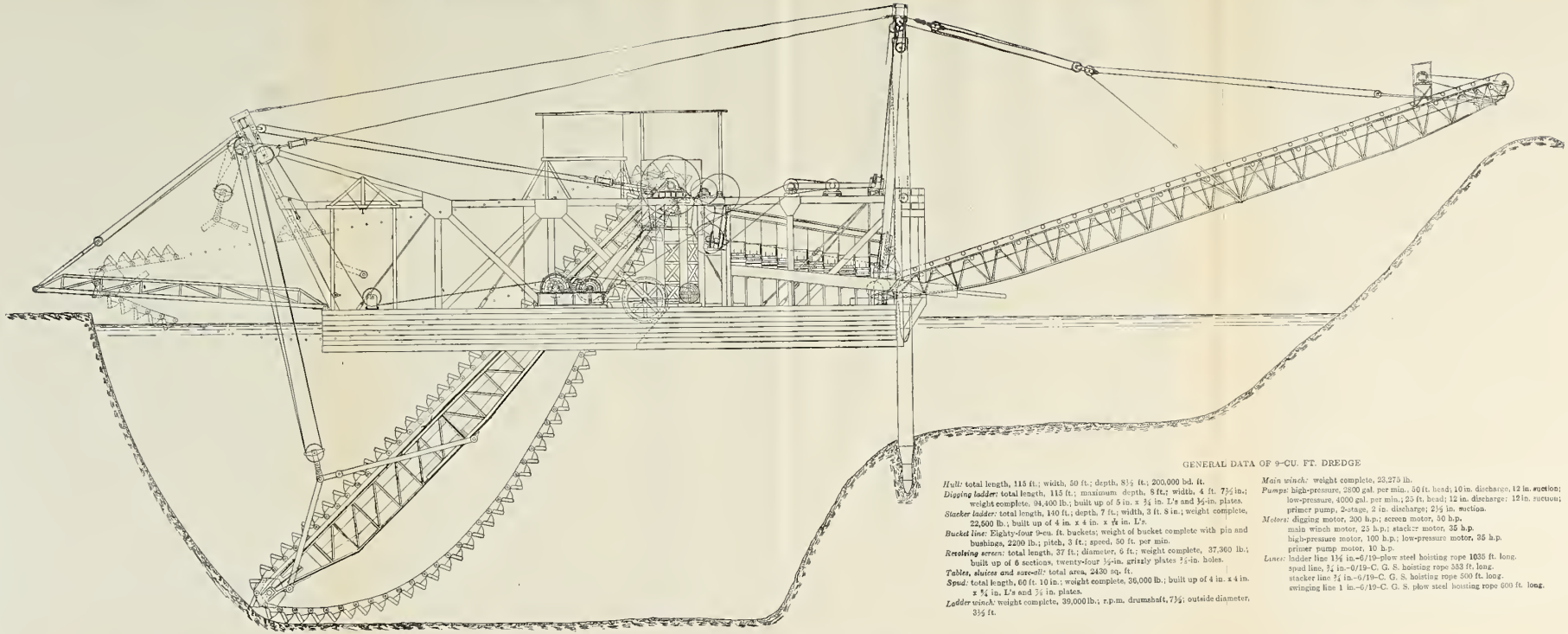


FIG. 4 PLAN, 9-CU. FT. DREDGE



GENERAL DATA OF 9-CU. FT. DREDGE

Hull: total length, 115 ft.; width, 50 ft.; depth, 8½ ft.; 200,000 bd. ft.

Digging ladder: total length, 115 ft.; maximum depth, 8 ft.; width, 4 ft. 7½ in.; weight complete, 94,400 lb.; built up of 5 in. x ¾ in. L's and ⅜-in. plates.

Stacker ladder: total length, 140 ft.; depth, 7 ft.; width, 3 ft. 8 in.; weight complete, 22,500 lb.; built up of 4 in. x 4 in. x ⅝ in. L's.

Bucket line: Eighty-four 9-cu. ft. buckets; weight of bucket complete with pin and bushings, 2200 lb.; pitch, 3 ft.; speed, 50 ft. per min.

Revolving screen: total length, 37 ft.; diameter, 6 ft.; weight complete, 37,360 lb.; built up of 6 sections, twenty-four ½-in. grizzly plates ⅝-in. holes.

Tables, sluices and sure-all: total area, 2430 sq. ft.

Spud: total length, 60 ft. 10 in.; weight complete, 36,000 lb.; built up of 4 in. x 4 in. x ¼ in. L's and ⅜ in. plates.

Ladder winch: weight complete, 39,000 lb.; r.p.m. drumshaft, 7½; outside diameter, 3½ ft.

Main winch: weight complete, 23,275 lb.

Pumps: high-pressure, 2800 gal. per min., 50 ft. head; 10 in. discharge, 12 in. suction; low-pressure, 4000 gal. per min., 25 ft. head; 12 in. discharge; 12 in. suction; primer pump, 2-stage, 2 in. discharge; 2½ in. suction.

Motors: digging motor, 200 h.p.; screen motor, 50 h.p.
main winch motor, 25 h.p.; stacker motor, 35 h.p.
high-pressure motor, 100 h.p.; low-pressure motor, 35 h.p.
primer pump motor, 10 h.p.

Lines: ladder line 1¼ in.–6/19–plow steel hoisting rope 1035 ft. long.
spud line, ¾ in.–6/19–C. G. S. hoisting rope 353 ft. long.
stacker line ¾ in.–6/19–C. G. S. hoisting rope 500 ft. long.
swinging line 1 in.–6/19–C. G. S. plow steel hoisting rope 600 ft. long.

FIG. 5 ELEVATION, 9-CU. FT. DREDGE

California type, and with its later modifications is the one to receive the most attention in this paper, and unless otherwise stated the descriptions are for this type of dredge. A plan and elevation of this type are shown in Figs. 4 and 5.

12 The original New Zealand type slightly modified is still sometimes used for the smaller, shallower and richer areas, where it is said to do very good work. It is cheaper to build and under certain conditions may prove a better investment than the more expensive California type.

13 The earlier dredges used buckets holding 3 cu. ft., then the 5-ft. size became numerous (Fig. 6), and later the 7, 8 and 9-ft. sizes (Fig. 7) were much used. Many went back to the revolving screen,

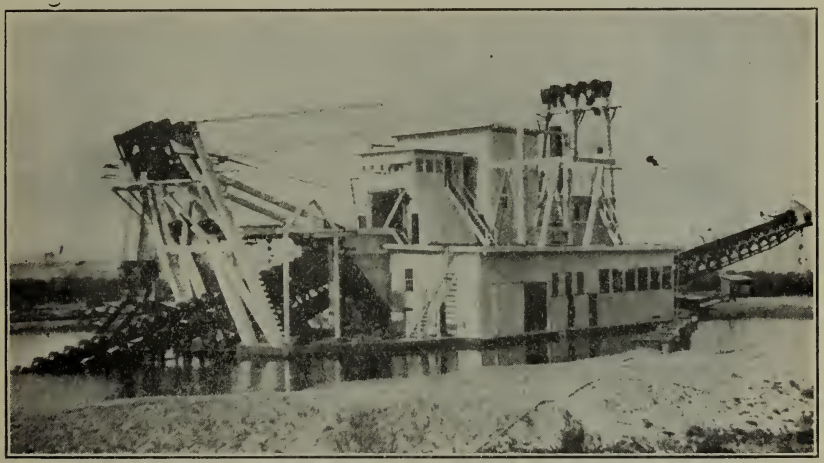


FIG 7 MODERN 9-CU. FT. GOLD DREDGE, 1911

noticeably in localities where much clay existed. In 1905 a 13½-ft. dredge was built near Folsom, Cal. (Fig. 8), and since that time several with 13½ to 16-ft. buckets have been put in operation (Fig. 9).

14 Having gained a general idea of the development of this type of gold dredge, its different component parts will be taken up in detail.

BUCKET

15 The bucket proper is made up of base, hood and lip (Fig. 10).

16 *Base.* The base is made of cast steel, many grades being used. It should be hard so as to stand wear, tough so as to stand the strain of digging, and should not be too expensive, since it has constantly to be replaced when worn or broken. All dredge costs

are figured on a basis of cents per cubic yard of gravel handled, so that the first cost is not vital so long as it will handle the requisite number of cubic yards before giving out.

17 The most common form of base has one hub or back eye fitting between the two front eyes of the bucket behind it. Some use two back and three front eyes. The advantage of the three-eye bucket is that a smaller diameter pin may be used, there being four planes of shear instead of two; the disadvantage is that the cost of machining is greater and there seems to be a tendency to crack these buckets between the two back eyes, owing probably to a prying

FIG. 8 MODERN $13\frac{1}{2}$-CU. FT. GOLD DREDGE, 1905

action under hard digging conditions, particularly after the pins are partly worn.

18 Besides the ordinary high-carbon open-hearth steel, chrome nickel and manganese steels are used. One essential is that the bucket casting should be thoroughly annealed before it is machined. It is my opinion that many breaks in bucket bases have been caused solely by improper annealing.

19 A template having long guiding bushed holes is useful in drilling for the hood so that the rivet holes will be accurately spaced both inside and out. This is essential in order that there may be no lost time in making replacements. In machining the base the holes in the eyes should be very carefully bored so as to be exactly parallel with the hole in the hub. In order to do this the bucket may be fitted with its bushing, a shaft put through the hub, the bucket

brought tightly against it, and then the eyes bored parallel to this shaft.

20 *Hood.* The bucket hood is made either of pressed steel or cast steel. There are also buckets used which have the hood cast as an integral part of the base. The pressed steel hoods are made in either one or three pieces. The thickness depends on the size of bucket and the nature of the digging, and varies from $\frac{1}{4}$ in. to $\frac{1}{2}$ in. It is sometimes reinforced at the back where the greatest wear occurs. The cast-steel hoods are usually made about $\frac{3}{4}$ in. thick, have a stiffening rib running up the middle, and sometimes shoulders for the lip to rest on and a shoulder for the base to fit into.

21 The advantages of pressed-over cast-steel hoods are their cheapness, lightness, and ease of repair if bent or broken. A disad-

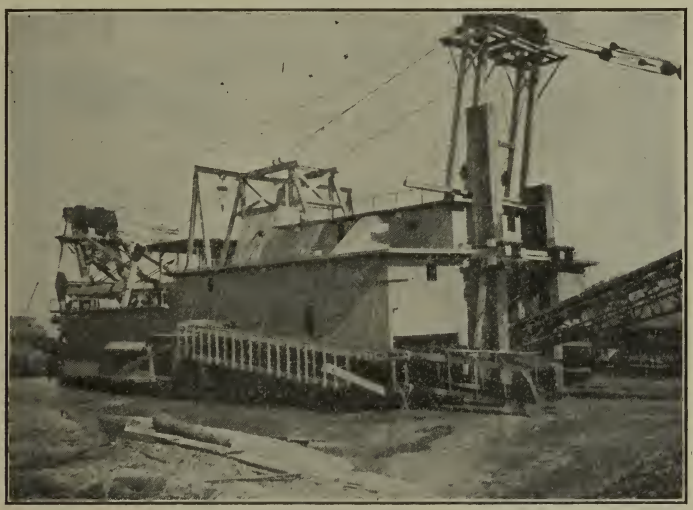


FIG. 9 MODERN 16-CU. FT. GOLD DREDGE, 1911

vantage is that the strength cannot always be put where wanted except at extra cost.

22 A standard drilled base may be used as a template for drilling the hood. The author considers the one-piece preferable to the three-piece hood because (*a*) it will stay water tight; and (*b*) there are fewer rivets to work loose and replace. The disadvantage is the difficulty of pressing a single piece into the proper shape, making the cost slightly greater than if three pieces were used.

23 *Lip.* Bucket lips are now nearly all made of cast manganese

steel. At one time the author used a very hard high-carbon nickel-steel forging, its only advantage being that it could be shaped on the ground to fit several different hoods and bases which it was desired to use up. A record was kept of about 100 of these lips and it was found that for cost and life they were very close to the manganese steel. For use in out of the way places and in emergencies they do

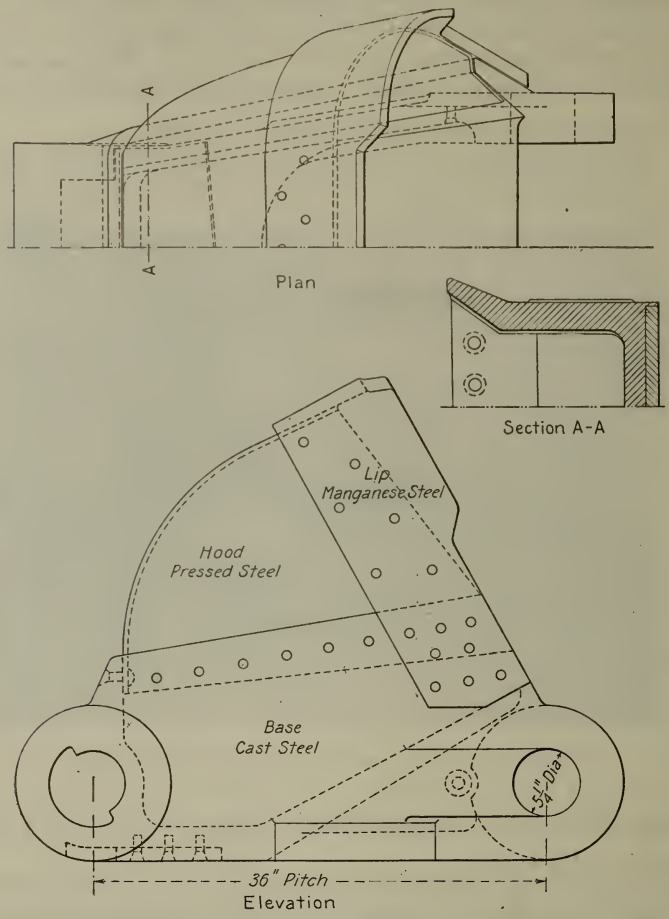


FIG. 10 9-CU. FT. BUCKET

very well. Their greatest disadvantage is that they cannot be readily forged into the more intricate shapes now commonly used.

24 The lips in use vary considerably in design according to the nature of the ground and the fancy or judgment of the designer.

In the first place, a lip must be made sturdy enough to perform its work without bending or breaking, even up to the time it is ready to be replaced. If it is bent or broken before this time, damage is done the hood, with the resultant extra cost of repairing the hood as well as replacing the lip. The thickness of the lip is determined by the above conditions, but its width is a matter of judgment, since if made too wide the capacity of the bucket is materially reduced by the time it is worn out; if too narrow, it must be replaced often and a large loss made in each replacement both in time and unused metal.

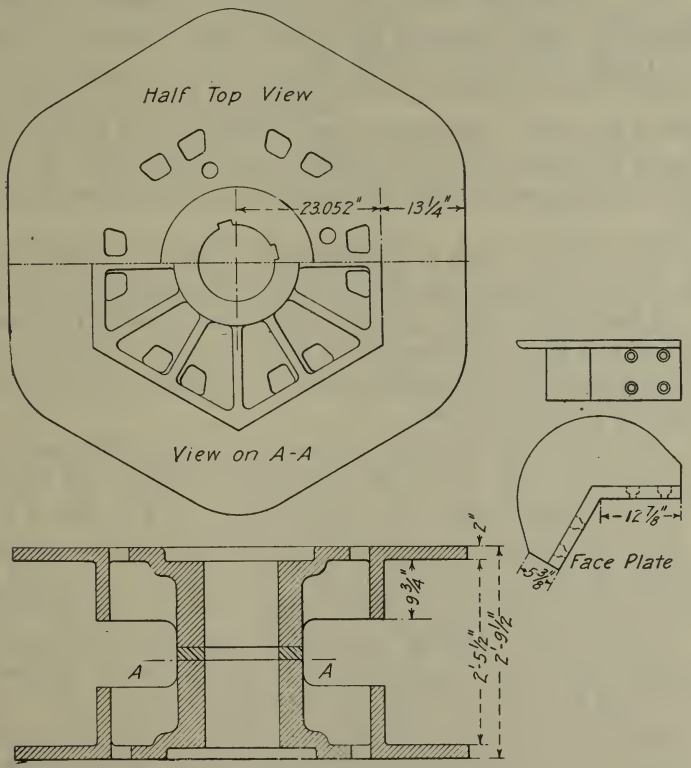


FIG. 11 LOWER TUMBLER FOR 9-CU. FT. DREDGE, SHOWING WEARING PLATE

25 Some engineers prefer a highly arched digging face, others a rather flat face with sharply rounded corners. Those favoring the arched face contend with good reason that it is a stronger shape, and those favoring the flat face and rounded corners think that this shape is a better digger. The shape of the digging edge is another matter in which designers differ; some make it perfectly straight, others

prefer corner extensions at the points where the wear indicates most of the digging is done. The author is in favor of the corner extensions, since he believes they dig better and the lip is more completely worn out when it is ready to be replaced.

26 The fewer rivets used the smaller will be the cost of re-assembling, but care must be taken to have enough rivets to do the work without shearing or pulling out. On many of the lip designs a shoulder is cast so that the hood edge fits against it, thus taking much of the strain off the rivets.

27 *Pin.* There have been many attempts to improve the design of bucket pins, those ordinarily used being rather expensive to make. It is a round shaft with a flat lug or "L" at one end. The L fits into a slot machined in the bucket base and prevents the pin turning in the eyes. Dowel pins, keys, squared and slotted ends, tee shapes and many other devices have been tried, but so far as the author is aware none of them has proved as satisfactory as the old L design.

28 The pins are made of various kinds of steel, the object being to use a very hard material which is also tough and not too expensive. A high-carbon steel, low in sulphur and phosphorus, forged, machined, and then oil tempered, is most common. Various forms of nickel, chrome and manganese steel are also used, but so far have not become general practice.

29 *Bushing.* The bushings are made in pairs, half round, and fit a recess in the bucket hub, placed at such an angle that the wear will be evenly distributed. They should be a snug fit, but $\frac{1}{4}$ in. to $\frac{1}{2}$ in. short on the ends to allow for the flow of the metal. In using manganese steel this tendency to flow must always be taken into consideration and suitable provision made for it in the dimensions.

30 *Insert Plates.* During the last few years manganese steel insert plates have come into general use in order to protect the bottom of the bucket base near the hub, at which point excessive wear occurs on account of the slip on the upper tumbler. Often buckets are found to be worn out at this point but perfectly good everywhere else. The manganese plates are of such a shape that after they are put in place and the softer steel of the base peened over the edges they retain their position without other means of fastening.

LOWER TUMBLER

31 At the lower end of the digging ladder the buckets pass over a 6-sided flanged tumbler (Fig. 11) which revolves between forks at-

tached to the digging ladder. It is cast of open-hearth steel. Wearing plates are used to protect the faces and the flanges. The detail of design varies on different dredges. Some use a tumbler having curved extensions at the corners of the flanges, others have the corners rounded but not extended, and still others use a circular flange. The extended corners require less metal but are likely to be broken off, and the circular flange may interfere with the digging. The design in which a side view will show a hexagonal shape with rounded corners is strong and cannot well interfere with the digging. The height of flange is another point of difference and varies from 6 in. to 13 in. at the corners. It must be made high enough to keep the buckets from running off, but if too high may interfere with the digging. It is also expensive.

32 The slope or flare of the flange varies from 0 to 1 in 10. It is claimed by some that if the sides are made too straight that the buckets are likely to run off and if the flare is too great the eyes of the buckets will wear thin at the bottom. I do not know that any standard of flare has been adopted, but I am inclined to the belief that a straight or nearly straight flange is best, all things considered.

33 *Wearing Plates.* The flanges and tumbler faces are protected by wearing plates (Fig. 11) made of some hard steel, nickel, chrome and manganese all being used. So far as wearing qualities are concerned the manganese steel gives the best result, but this material is so hard that it cannot be machined and it is difficult to make the plates fit the tumbler accurately; also, since the rivet holes have to be cored they are likely to be a bad fit. Besides these disadvantages the manganese steel plates are very destructive to the bucket line and it is better to wear out tumbler plates than buckets. Several tumblers have recently been made with a manganese steel disk set into a recess at each corner and held in place by peening over the edges in a manner similar to the bucket insert plates described above. This method is said to have given very good results. Some use separate plates for protecting faces and flanges, while others have the wearing plates made so as to protect both faces and flanges. The advantage of the last method is that the plates are very strong and not likely to break or work loose.

34 *Shaft.* The lower tumbler shaft is usually made of nickel steel, either oil tempered, annealed, or both, and may be hollow bored or solid. It should be very securely keyed to the tumbler, and the author has found that four taper keys, two at each side, 90 or 120 deg. apart, have given the best results. Provision should be made for

driving the keys out from the inside and the tumbler should be either shrunk or pressed on.

LADDER ROLLER

35 The ladder rollers (Fig. 12) serve to support the buckets on their way from the lower to the upper tumbler. Cast iron was formerly used, but now cast steel, either high carbon, chrome or manganese, is most in favor. Very hard chilled white iron cast around a soft core has given very good results; also manganese steel made in the same way has proved even more efficient so far as wear is concerned. The writer's objection to the manganese roller is that after considerable use the soft core, if made thin, is likely to become

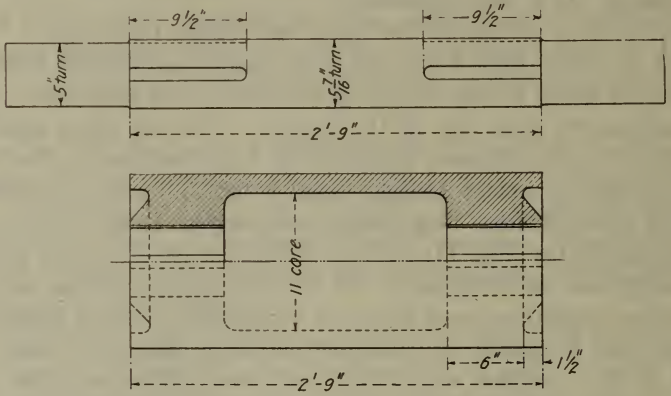


Fig. 12 Ladder Roller for 9-Cu. Ft. Dredge

loose, and if made thick the difference in expansion of the two metals will cause the outside shell to crack.

36 A high-carbon or chrome-steel casting makes a good roller, and can be made just soft enough to machine; it will then wear well and need not have a soft iron core. Particularly good results have been secured by the author with chrome-steel rollers made in this way.

37 *Shaft.* There has always been trouble in keeping shafts tight in the rollers. The writer uses two taper keys at each end, 90 deg. apart, the same plan as for the lower tumbler shaft. This method of fastening has proved perfectly satisfactory.

UPPER TUMBLER

38 There have been many forms of upper tumbler tried, but now

the common practice is to use a 6-sided plain casting, onto which is bolted a set of heavy cushion plates with guides cast on the sides (Fig. 13).

39 *Cushion Plates.* These plates are made to lap over the corners and fit into transverse slots in the tumbler casting. This is done to prevent them from becoming loose, as was the case before this design was perfected. They are made of cast steel, either high carbon, chrome, nickel, or manganese. The cushion plates are bolted to the main tumbler casting by means of large bolts $1\frac{1}{2}$ or 2 in. in diameter with square countersunk heads on the outside (Fig. 14).

40 *Wearing Plates.* Wearing plates are used on the cushion

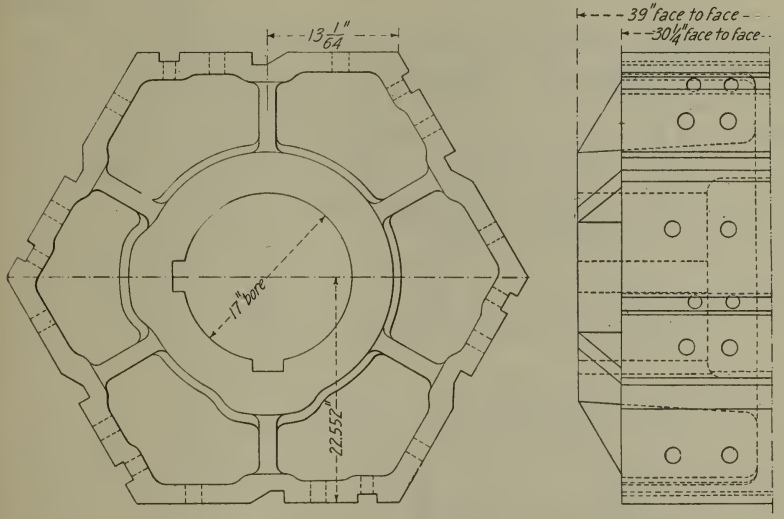


FIG. 13 UPPER TUMBLER FOR 9-CU. FT. DREDGE

plates, both for the faces and the guide flanges on the sides. These plates may be made of any material desired, but the same objection to manganese exists on the upper as the lower tumbler. The author uses plates forged from worn-out bucket pins and then gives them an oil temper. These have given good results, but care should be taken not to temper them too hard, since they then become brittle and will give poor service. The wearing plates are riveted with countersunk rivets to the cushion plates. Riveting is better than bolting for this purpose.

41 *Shaft.* It is imperative that the upper tumbler shaft should be of ample strength and made of excellent material since it has

to bear the brunt of the digging. A factor of safety of not less than 10 is recommended, and it would do no harm to have it even higher. It is customary to use a material of high tensile strength, since if made of low tensile strength material the shaft will have to be so large that it will interfere with the design of the upper tumbler which it carries. Nickel steel, hollow bored, oil tempered and annealed shafts are much used, having a tensile strength of 90,000 lb., elastic limit 60,000, elongation 22 per cent, contraction 50 per cent. Excellent results have also been obtained with Krupp crucible steel shafts, hollow bored, oil tempered and annealed, having a tensile

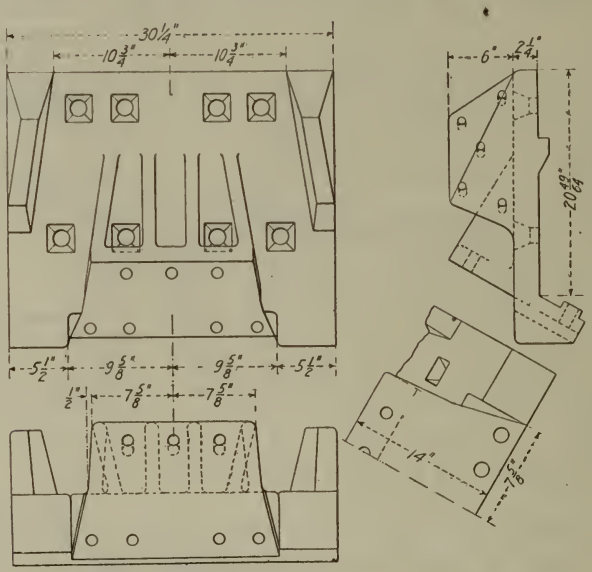


Fig. 14 Cushion Plates for Upper Tumbler, 9-Cu. Ft. Dredge

strength of 80,000 lb., elastic limit, 43,000, elongation 22 per cent, contraction 45 per cent. Lately the author has used a nickel steel solid shaft which has been thoroughly annealed but not oil tempered. This has given excellent results and is inexpensive. The physical properties are as follows: Tensile strength, 80,000, elastic limit 50,000, elongation 25 per cent, contraction 45 per cent. A shaft of soft iron or steel, having a low tensile strength, but great toughness, such as Lloyds' test steel, would be the ideal shaft for this purpose were it not for the necessity of making them so large that there is no

room left to fasten the cushion plates to the upper tumbler. By using a 7 or 8-sided tumbler this difficulty could be overcome.

DIGGING LADDER

42 The design of the digging ladder varies with the depth and nature of the ground to be handled. It is either a plate girder or a lattice girder or truss (Fig. 15).

43 The plate girder type is the most common and in the writer's opinion the best. A little weight may be saved by using the lattice type, and although it may be as strong as a plate girder, there is in the lattice type more danger of rivets working loose, causing ultimate collapse.

44 Ample space should be allowed under the ladder rollers, otherwise the spill from the buckets will collect and prevent the rollers turning. The ladders are made in several sections so that they may be more easily transported. Where the splices occur great care should be taken to make the joints of ample strength. The author has seen several digging ladders fail and in nearly every case the failure occurred at one of the splices. In figuring the strains in a digging ladder it should be considered not only as a beam supported at two or three points, but also as a column. Where a single line is used for side swinging, the strains must also be figured for side as well as vertical stress.

45 *Lower Forks.* At the lower end of the ladder cast-steel forks are riveted to the structural portion of the ladder. These forks are made differently by various designers, but their duty in all cases is to carry the lower tumbler bearings and give a space for the lower tumbler to revolve in. These forks should have great lateral strength since there is a tendency to spread under the severe digging conditions. The writer has found it of advantage to make the inside edges of these forks fit very closely to the outside of the tumbler, since if a large space is left rocks lodge and grind out both the tumbler and the forks. In his last design he has put in replaceable manganese steel liners on the inside of the forks and hopes that it will prevent much of the wear on tumbler flanges.

46 *Lower Tumbler Bearings.* There are several designs of lower tumbler bearings used. In the most common the lower portion of the forks are made to form one-half of the bearings and to this the other half is bolted. In this design a replaceable cast-iron bushing is made to fit inside the main casting, often of the self-aligning type. A design similar to that of the locomotive connecting-rod end, which

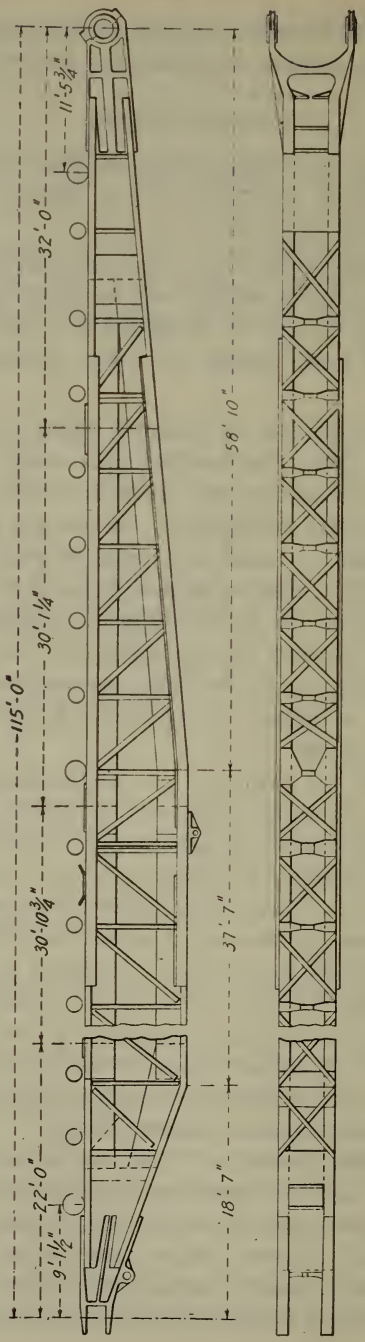


FIG. 15 DIGGING LADDER FOR 9-CU. FT. DREDGE

is fastened and adjusted by means of taper keys, has been used by the author. The bearing is made of cast iron in two halves and slides between the upper and lower extensions of the forks. Square holes are machined through these extensions to take the keys. The half bearings are interchangeable and can be used for the upper or lower half and for either side. This style of bearing has been found very satisfactory, since it is cheap and easily replaceable.

47 Many attempts have been made to use protective devices for keeping grit out of the bearings, rubber, felt, brass and iron rings having been tried. The rubber and felt have been entirely discarded. Some still use bronze or iron protective rings, but they are all more

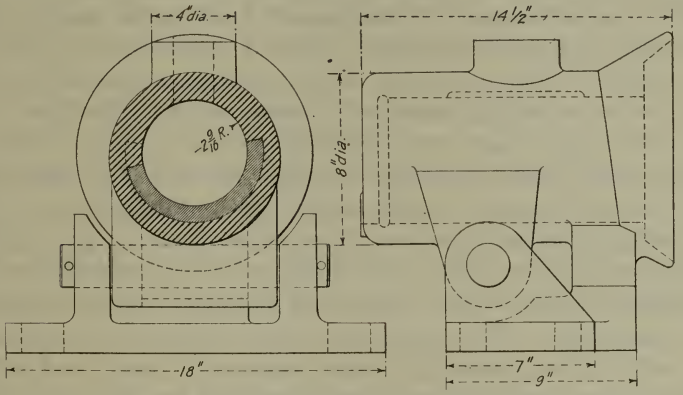


Fig. 16 Ladder Roller Bearing for 9-Cu. Ft. Dredge

bother and cause more delays than they are worth. A hole through which grease may be introduced is provided at an easily accessible place on the bearings. The grease is introduced by means of a grease gun with a screw-operated piston.

48 The writer uses a lower tumbler shaft with shallow flanges turned at either end. These flanges fit into recesses on the outside of the tumbler bearings, thus preventing the ladder forks from spreading. Where hollow shafts were used it was common practice at one time to put a rod lengthwise through the shaft, a nut at either end preventing the forks from springing apart. With the modern practice of making the lower forks extremely massive there is little danger of springing and precautions need not be taken in this direction.

49 *Roller Bearings.* Spaced along the digging ladder at regular intervals are the roller bearings supported in chairs by pins which

allow movement in a plane at right angles to the length of the ladder (Fig. 16). The first bearings used were fixed rigidly and much trouble was experienced by breaking roller shafts, probably caused by movement in the ladder truss. I think it likely, with the heavier ladder construction now in vogue and the larger-sized roller shafts, that the fixed bearing would give little trouble; but even so, the present common design is better since the bearing may be very quickly and easily removed, and this is a desirable feature in any part of a dredge.

50 The roller bearings are made with half-round bushings similar to those used in the bucket hubs. Cast iron, manganese steel and bronze are used for this purpose. Cast-iron roller bearings are being replaced by cast-steel. Where a self-adjusting bearing is used it should be supported near the middle and be of ample length, otherwise it will tend to tip down and wear out unevenly and rapidly.

51 *Catch-all Pan.* In some of the dredges a pan is provided below the rollers which conveys the slop-over of the buckets to the lower end of the ladder where it will be picked up again. This is a very good idea, but care should be taken to make it at least 12 in. below the rollers and in small, easily replaceable sections. If not deep enough it is likely to be a source of much annoyance by filling up and stopping the rollers, or if made in long unprotected sections it will wear out and require much work and loss of time in replacing.

52 *Suspension.* The lower end of the ladder is supported at either one or two points, the single suspension being used for the shorter ladders and the two-point suspension for the longer ones. Several different designs are in use, the most common being a double set of blocks, one on each side, attached to heavy forged rods or bars connecting with the ladder. Heavy chains, forged or cast links and pin links are also used. Another type has a single set of blocks attached to a cross-piece, the connection to the ladder being made from each of its ends. The attachment to the ladder is usually made by means of a cross-shaft or pin running through bearings fastened to the under side of the ladder.

53 *Upper Forks.* The upper end of the digging ladder is pivoted either on the tumbler shaft-bearing support or on a separate mounting entirely detached from the bucket drive. By employing the first method the tumbler shaft may be used as a center and the relative position of the bucket line and upper tumbler will remain constant for all depths of digging. In the second method the pull on the bucket line will be at different angles according to the position of the ladder. The first method, therefore, has a decided advantage, but

it also possesses the disadvantage of preventing the easy removal of the upper tumbler and shaft, is more expensive, and in case of accident to the ladder the bucket drive may be involved through the tumbler shaft-bearing support. Massive cast-steel forks are used when the ladder is pivoted on the tumbler-bearing supports, and they should extend well down the structural part of the ladder in order to give ample space for riveting.

54 *Chafing Beams.* Along the sides of the ladder wooden chafing beams are bolted, the outer faces of which are protected by steel plates. The author prefers beams put at both top and bottom of the ladder so that they will steady it when digging hard. The beams on the ladder rub against similar ones on the sides of the well. Those in the well are put either at right angles to the ladder at intervals, or one may be placed on each side of the well parallel to the deck and on the streak above water line. If this form of chafing beam is used both the ladder and the well beams should have beveled timbers fitted above them so that rocks will not lodge and cause trouble when the ladder is raised or lowered. The disadvantage of placing the chafing beams at right angles to the ladder is that in case anything catches on one of them near the bottom and tears off a plate it is very difficult to repair, while if the beams are entirely above the water line they are easily accessible. The only disadvantage of placing them in this way is that the distance from the contact point of ladder and beam to the ladder's end is greater by the draft of the hull, and in case of side strain the bending moment is greater. Where a shore sheave is used and the end of the swinging line is attached to the lower end of the ladder there is very little side strain, and in this case the horizontal wearing plates are best, in the writer's opinion. Where a single swinging line is used, running directly to the shore, considerable side strain is exerted on the ladder, and the chafing beams at right angles to the ladder are preferable. Some designers provide rollers for the sides of the ladder, which serve the same purpose as the chafing beams.

BUCKET DRIVE

55 The buckets are driven by a chain of gears mounted on the main gantry. The use of a chain or shaft drive from the lower deck has now been entirely abandoned in California practice. Some dredges drive direct from the digging motor to the pulley shaft on the main gantry. Others use a countershaft on the main deck, from which also is driven the ladder hoist. Neither type has any great

advantage over the other, and if properly designed either method will give satisfactory results.

56 The upper tumbler may be driven by a gear at one end of the tumbler shaft, or one at each end, with a double set of intermediate pinions, gears, and pulley-shaft pinions. The single drive is of course cheaper and simpler, but experience has shown that after many years of service the single drive is inclined to get out of line, even

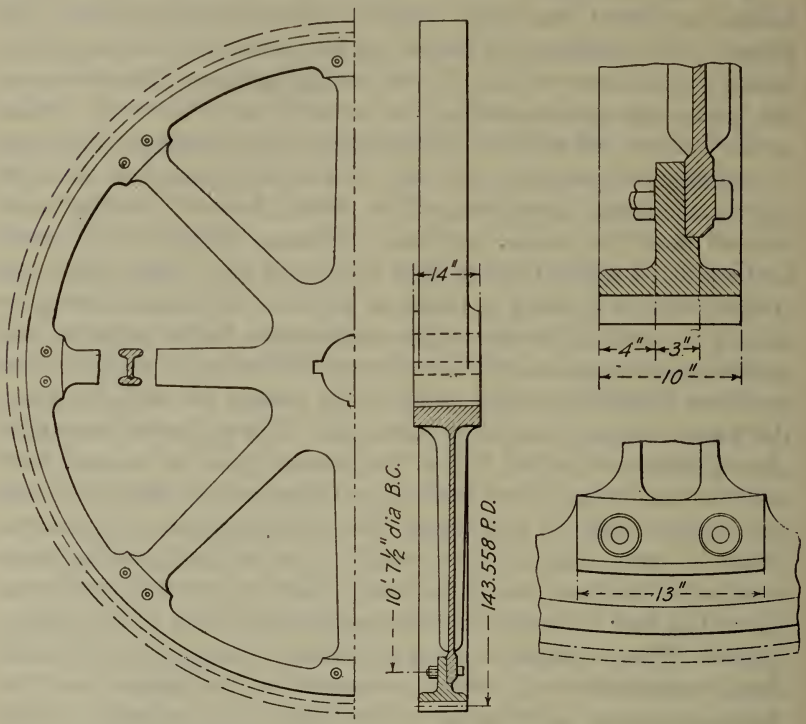


Fig. 17 Bull Wheel for 9-Cu. Ft. Dredge

when very solidly built. The writer very much favors the double drive and would advise its use.

57 *Bull Wheel.* The bull wheels or tumbler-shaft gears are cast of steel in sections, sometimes with a split and sometimes with a solid hub (Fig. 17). The rim is cast in one piece or in halves. Some designers use cut teeth, others rough cast. There can be no doubt but that the cut teeth run more smoothly, but for the rough work in hand the writer does not believe that the extra cost is warranted,

particularly since the teeth are soon worn down and the benefit of the cut teeth disappears. In order to prevent breaking, all sharp angles should be avoided in the design of the spokes. They are subjected to constant shocks and vibration, and the gears will invariably break at a sharp re-entrant angle should such occur at a vulnerable point. In order to make a snug fit and also to keep the rim from becoming elliptical, the spokes should either have a shoulder fitting on a machined inner edge of the rim, or the ends of the spokes should be turned to fit a machined shoulder on the rim. The latter method is preferable since it does away with sharp corners which, as explained above, are always points of weakness.

58 It has been common practice to use one large bolt to fasten each spoke to the rim. Two smaller bolts are better since a more even distribution of spoke cross-section can be maintained by this design and it gives a longer spoke end to bear against the machined rim. A good many spokes have been broken near the rim where the cross-section is ample but where a point of weakness exists owing to the re-entrant shoulder above mentioned.

59 *Intermediate Shaft Pinions.* The intermediate shaft pinions are made solid and as small in diameter as is consistent with good design. These pinions wear out quite rapidly and should be so arranged as to be easily replaced.

60 *Intermediate Shaft Gears.* The intermediate shaft gears are designed of a size to give the bucket line the desired speed. They are often made with separate rims so that they may be replaced when worn without throwing away the rest of the gear.

61 *Pulley-Shaft Pinion.* The pulley-shaft pinions are made similar to the intermediate shaft pinions and as small as good design will allow.

62 *Clutch.* Whether the digging motor is belted direct to the bucket-drive pulley shaft, or to a countershaft on the ladder-hoist foundations and from there to the bucket-drive pulley shaft, some form of friction clutch is used to connect and disconnect the bucket drive, and for purposes of safety in case the buckets encounter obstructions beyond their strength. Any of the standard forms of friction clutches may be used, but the author prefers the electromagnet clutch and uses it in all his designs. It is by far the simplest to operate, is positive and will slip at exactly the same pull every time according to the amount of current sent through its coil by the rheostat controlled from the pilot house.

63 The clutch used is very simple in design (Fig. 18). A cast-

iron pulley is made so that there is a flat ring or web joining the spokes at about half the distance from hub to face. To this web is bolted a soft cast-iron ring. Between this and the pulley are inserted two thin sheets of copper and one of soft iron between the copper plates. These ring-shaped sheets are used in order to keep the lines of force where they are most wanted and to prevent the current making a permanent magnet of the pulley, shaft, etc., which would cause the clutch to drag when released. The pulley has a long hub

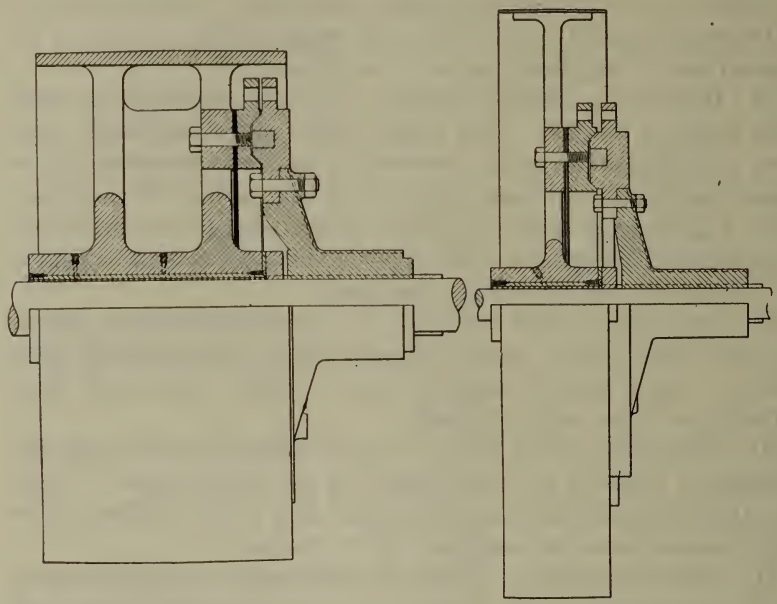


FIG. 18 MAGNETIC CLUTCHES FOR BUCKET DRIVE AND MAIN WINCH, 9-CU. FT DREDGE

bushed with bronze and runs free on the shaft. A long cast-iron hub with two feather keyways at 180 deg. and a web extending out to the same diameter as the ring on the pulley is made to slide along the shaft. Onto this web is bolted another soft iron ring similar to the one bolted to the pulley. An annular groove of rectangular cross-section is machined in this ring midway between its inner and outer circumference into which fits a coil of magnet wire. By introducing an electric current into this coil through controller rings, magnetic lines of force are induced which will draw the two soft iron rings together, the pressure depending on the amount of current put through and the number of turns in the coil. The edges of the soft iron rings

are beveled so as to make a female joint on the pulley and a male joint on the sliding hub. The steeper the slope of these edges the less current will be required to do the requisite work, but if the cones are made too steep it will be difficult to disconnect the clutch. Forty-five degrees does very well. Good results are given by 440 turns of No. 16 magnet wire, and 4 amperes at 110 volts are put through the coil when working average ground. These figures are for a 200-h.p. clutch.

64 *Shafts.* The shafting for the bucket drive should be made of ample dimensions and of good material. Nickel steel annealed and a factor of safety of 10 are used by the author. The gears should be fitted very neatly and securely keyed.

65 *Bearings.* The upper tumbler-shaft bearings are made on an angle so as to have the bearing cap at right angles to the average line of pull of the bucket line. Cast-iron liners are sometimes used, but the author prefers a babbited bearing. The bearings should be well fastened so that they cannot slide forward, there being a very heavy pull in this direction. These bearings, like the others of the bucket drive, are made of cast steel.

66 The intermediate and pulley-shaft bearings are made of the ordinary type of babbited base and cap, but are heavier than standard and use very large diameter bolts, which should be made of a soft grade of iron or steel, since unless this is done they are likely to crystallize and break.

67 *Brake.* It is customary to have the pulley shaft extended and a brake wheel keyed on. This brake is operated from the pilot house and is used by the winchman to hold the buckets from going over the upper tumbler when the bucket line breaks, to stop quickly in case of an accident, and to hold the bucket line when making repairs or changing buckets.

68 *Pulley.* Standard medart pulleys are generally used. They give considerable trouble on account of broken spokes. The author has recently used with success a wooden bucket-drive pulley similar to that commonly employed in stamp mills, and on the last boat he designed a medart pulley with spokes made extra heavy near the hub and tapering to a little more than standard cross-section at the rim, since nearly all the trouble arose from spokes breaking near the hub. If a cast-steel hub and spokes could be made it would be the best pulley for the purpose. Several of the steel foundries say that they would not care to attempt to cast a spider with as small spokes as the standard medart and that a very heavy spoke would have to be

made in order to overcome shrinkage trouble. If spokes could be made of cast steel and of as light or lighter design than the standard medart it seems as if it would be an ideal pulley for dredge use.

69 *Foundations.* Too much attention cannot be given to making the supports for the bucket drive of very sturdy construction. In the first place these foundations are 20 or 30 ft. above the main deck and are subjected to constant and racking strains. If any portion of

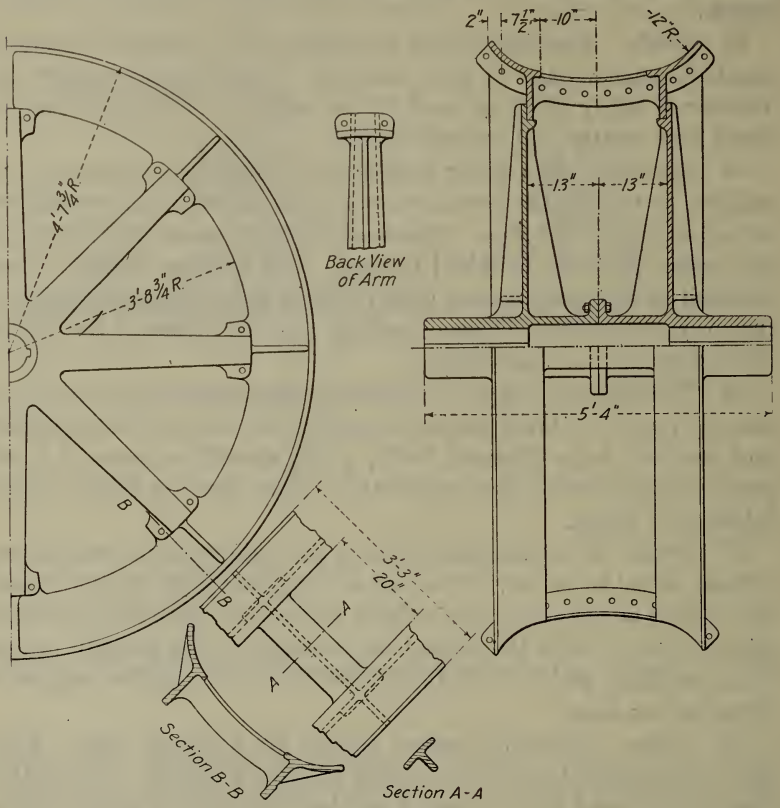


FIG. 19 BUCKET IDLER FOR 9-CU. FT. DREDGE

these supports gives it throws the shafting out of line with obvious results. One form of support now much used and considered very good has the upper cord of the main truss made high enough to take the upper tumbler-bearing supports directly on its top. By this method the main gantry is braced fore and aft in a most excellent manner and it is a much more rigid construction than where the

gantry extends above the cord. Large steel castings are used to tie the timbers together and to support the bearings directly. The details vary on different dredges and apparently have all given good service. One point which should be carefully looked into when designing this part of the dredge is so to place all bolts that they may be readily accessible and replaceable.

BUCKET IDLER

70 Where the depth of gravel to be dug demands a long string of buckets, an idler (Fig. 19) is used to support the sag. This idler is located at the throat of the well as low down as convenient. It is made up of a cast-steel hub and a double set of spokes bolted onto a wide rim made to fit the curve of the bucket lip. The shaft is keyed to the hub and revolves in two bronze bushed bearings set on the deck or other convenient place.

CATCH-ALL

71 Below the dump hopper an inclined set of grizzly bars is put in. These extend as far forward as the sag of the buckets will allow. The bars are put $\frac{3}{4}$ to 2 in. apart. Below this grizzly is a collecting pan which directs the gravel to the forward end of a sluice with riffles, occupying the space between the gantry posts and sloping toward the after portion of the dredge. This sluice is run back as far as the grade will permit and then dumps its contents into riffled side sluices running forward one on each side of the well. These discharge into the well 15 or 20 ft. forward of the well throat. Water is furnished from one of the pumps and washes the material dropping through the grizzly bars.

DUMP HOPPER

72 The hopper into which the buckets dump diverts the gravel into the screen. It is made either with a steel lining or by having a depression left at its bottom which fills up with rocks and gravel, thus forming its own lining. It is obvious that the last method is much more economical than the steel-lined hopper and that so far as its bottom is concerned it will last indefinitely; the only difficulty is that unless the hopper is left of ample dimensions it is likely to clog up when digging large quantities of material, particularly clay. By using a steel lining for the bottom of the hopper a little space may be saved, and if a large quantity of clay is to be handled the latter method might be preferable, but for ordinary conditions the rock bottom seems to be all that is necessary.

73 The hopper is built up on three sides of the upper tumbler, the top being at about the level of the bucket lips as they go over the tumbler. The forward upward edge is built to such a height as to

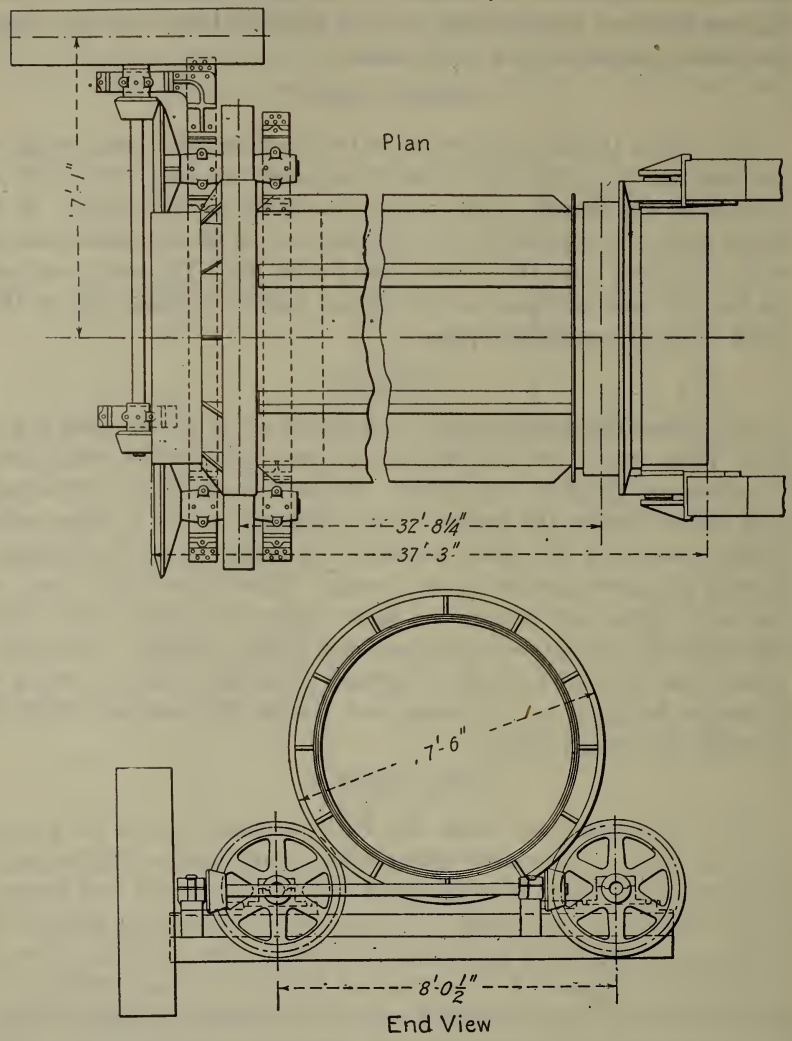


FIG. 20 REVOLVING SCREEN FOR 9-CU. FT. DREDGE

miss the bucket lips by a few inches when the ladder is in its lowest position, and it is of advantage to put inclined guide plates on each side of the bucket line to divert the slopover of the buckets into the

hopper below. The rear vertical wall of the hopper is subjected to considerable wear by the striking of the rocks and gravel against it when the buckets dump and it should be lined with an easily replaceable steel plate. A good way is to suspend a $\frac{1}{2}$-in. plate from the top by means of a few bolts, leaving the bottom entirely free.

SCREEN

74 Two types of screen are in common use, the revolving and the shaking. The first dredges used the revolving screen, then the shaking type was introduced at Oroville and for a time most of the dredges were built in this way. On the Yuba it was found that these did not break up the clay in a satisfactory manner and revolving screens were introduced. This type proved better for this district, and since then many have gone back to the revolving screen even in other districts. The object of a screen is not only to size the material passing through it, but also to disintegrate and wash it. This the revolving screen does more thoroughly than the shaking screen where clay or cemented gravel is present. The shaking screen gives a more even distribution of the fines, requires less power to operate, and its plates last longer and are more easily replaced. For easily washed material the writer prefers the shaking screen, but for gravel hard to wash, the revolving type is decidedly preferable.

75 *Revolving Screen.* The design of the revolving screen (Fig. 20) has been well worked out and very little trouble is experienced with this part of the dredge. Most of the delays in its operation are caused by the necessary replacement of worn plates. These plates should be put in so as to be readily removable and not too close a fit made; rather leave considerable space between them, since if this is not done they will spread and be hard to take out. Bolts with countersunk heads on the inside and split washers under the nuts on the outside are used in preference to rivets. They hold as well and can be removed more readily.

76 The screens are built up in sections usually with cast-steel ribs at intervals to retard the movement of the gravel. These steel ribs also serve as a framework, to which are fastened the perforated plates. The size of perforations depends on the conditions and varies from $\frac{3}{8}$ to $\frac{3}{4}$ in.

77 Around each end of the screen is a steel ring or tire preferably made in halves so as to be readily removable. Two drive rollers on which the ring at the upper end of the screen rests are used on most of the older dredges, but a single drive roller under the center with

two side guide rollers is coming into use on the newer boats. The rollers driving the screen are themselves driven by beveled gears and pinions. Owing to the difficulty of keeping grit out of these gears they are made of manganese steel, with a soft iron core cast in the hub. Rollers are used at the lower end to support the screen and a pair of manganese steel thrust rollers are provided to keep the screen from moving toward the stern. The bevel gears are driven from a countershaft above the screen, the driving motor also being located there. The countershaft is used in order to reduce the speed properly, so that excessively large gears or pulleys need not be employed.

78 At the lower end of the screen is a hopper which directs the coarse gravel onto a belt conveyor. In order to prevent clogging at this point the hopper should be placed on a grade of not less than 30 deg. and should be raised on the side on which the revolving screen moves upward, the gravel being carried up on this side of the screen.

79 *Shaking Screen.* The shaking screen (Fig. 21) is built with channel iron sides. The bottom is formed of angle iron cross-pieces, to which are bolted the perforated plates. The angles should extend beyond the sides so that the top of the channels may be braced from their ends. The screens are built in pairs and mounted tandem so as to balance each other, one moving forward when the other moves aft. They are suspended from the timbers above, some using adjustable, others non-adjustable rods. The upper and lower bearings for these rods should be made of ample size, since they wear surprisingly fast considering the small movement. The lower end of the after screen is constructed so as to direct the coarse material to the short discharge shoot onto the conveyor. This part of the screen is not perforated. The size of screen perforations depends on conditions and as in the revolving screen varies from $\frac{3}{8}$ to $\frac{3}{4}$ in.

80 The shaking screens are driven by means of four connecting rods, operated either by cranks or eccentrics on a thwartship shaft. The author is inclined to favor the eccentric drive owing to its larger wearing surface. If eccentrics are used they should have long hubs and double keyways, otherwise they are likely to work loose under the constant pounding. The drive shaft may be placed at any convenient point, but a central location between the two screens is perhaps the best. The bearings for this shaft are placed outside the screens, and as near as possible to the cranks or eccentrics, and should be long enough so that they will not beat out under the constant hammering. Buffer springs are sometimes used to take up the strain at the end of each stroke, but this practice is not common. The connecting rods are

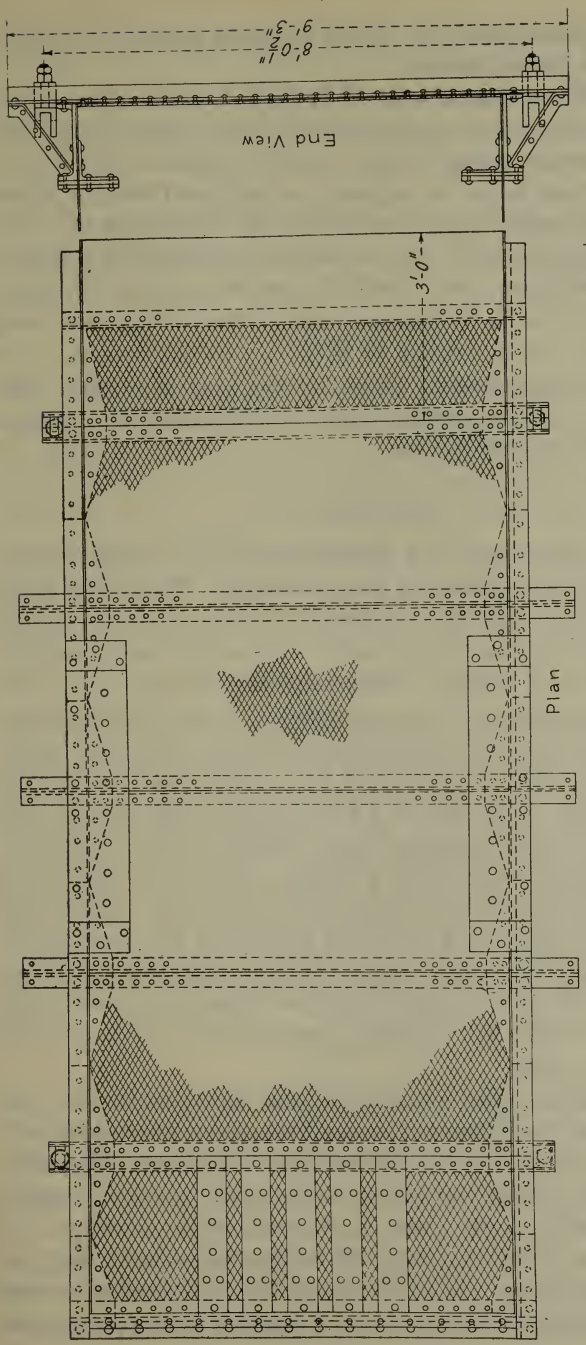


FIG. 21 UPPER SHAKING SCREEN, 7.5-CU. FT. DREDGE

usually made adjustable and any of the standard methods of fastening rod ends may be employed.

81 *Sprays.* The ordinary form of spray for washing the gravel as it passes down the screen is a perforated pipe or pipes. The water from the high-pressure pump is used, and the holes vary from $\frac{1}{4}$ to $\frac{3}{4}$ in. diameter. These holes are so placed as to direct the water where most needed. The spray pipes should have quick-opening valves at their lower ends so that they may be flushed out when the holes become clogged with roots, grass and other small particles of matter. Where much clay has to be washed some of the dredges are using large-sized nozzles set in groups at each end of the screen. It is thought that the large streams break up the clay in a more satisfactory manner than the greater number of smaller-sized streams furnished by the perforated pipes.

STACKER

82 *Ladder.* The design of a stacker ladder is a simple matter, since the loads and conditions are fairly constant. The lattice girder is the most common type and the strains in each member may be readily ascertained by graphic methods. The author uses a factor of safety of eight for this ladder. The lower end is supported near the stern of the boat and should be so designed as to allow the ladder to move up and down and also to allow the ladder to swing from side to side. The upper portion is supported by a bridle near the end and one near the middle. The ends of a wire rope are made fast to these bridles, its middle portion passing over a sheave which is in turn attached to the stacker ladder hoisting blocks. Instead of a single loop arrangement I prefer to fasten one end of the rope to the ladder hoist block, from thence passing it around a sheave at the upper bridle, then over another at the ladder hoist block, fastening the end to the middle bridle. This puts nearly double the supporting power at the upper end and prevents the buckling upward of the ladder when the dredge is surging badly.

83 *Conveyor Drive.* The common method of driving the belt conveyor is by means of a motor set on supports above the ladder 15 ft. or so back from its upper end. The motor is belted to a pulley shaft mounted near the end of the ladder, onto which is keyed a pinion with cut teeth which meshes into a cut gear on the conveyor belt pulley shaft. The author recommends a large size drive pulley covered with rubber belting. In order to make the belt contact greater an idler is placed near this pulley which brings the belt up and in con-

act with it for a large proportion of its circumference. By mounting the motor inside the ladder near the upper end a direct gear drive may be used, but this method has not been found as satisfactory as the belt drive described above. A silent chain drive is now being used a great deal. It allows the motor to be set nearer the drive and is said to work very well.

84 *Conveyor Belt.* Rubber conveyor belts are used and should have an extra thickness of rubber on the wearing side. The most important point is to select rubber of the very best quality.

DISTRIBUTOR

85 Below the screen is a hopper or distributor which is used to distribute the screened material onto the gold-saving tables. Here the ingenuity of the engineer may have full sway. The main object is to get the screened gravel evenly distributed on the gold-saving tables, and the way that it is accomplished makes very little difference. The distributor should be designed so that a certain amount of adjustment will be allowed to meet varying conditions. Where a shaking screen is used the problem is simpler than for a revolving screen.

GOLD-SAVING TABLES

86 This part of the dredge belongs more to the province of the mining than the mechanical engineer, so it will not be discussed at any length in this paper. Suffice it to say that the gold-saving tables should be specially designed to meet each different set of conditions. They occupy the greater portion of the dredge back of the main gantry and may be built either of plate steel or wood. The author prefers steel. The tables are covered with riffles made of angle iron or strips of wood covered with strap iron. The fine material is discharged either at the stern or some distance aft according to conditions.

LADDER HOIST

87 The ladder hoist (Fig. 22) is now usually driven by the digging motor from a countershaft located on the main deck. In a few cases it is driven from the bucket-drive pulley shaft on the main gantry. Owing to the load it has to carry it is made very heavy. A structural or cast-steel base is provided, on which are mounted the bearings. Single gearing is used and ordinarily the teeth are not cut. A friction clutch is provided on the pulley shaft to connect this part of the dredge machinery. From the standpoint of rope efficiency the drums

should be made much larger than is common practice, but excessively large drums are not warranted, unless the sheaves through which the same rope passes are made of corresponding dimensions. This entails

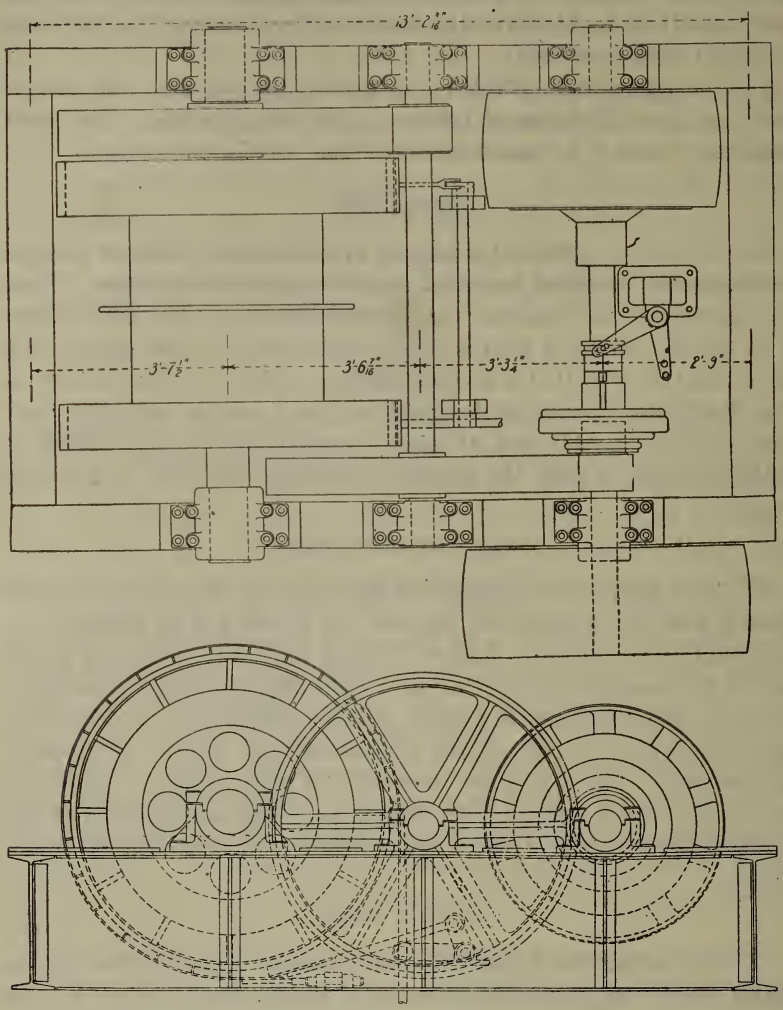


FIG. 22 LADDER HOIST FOR 9-CU. FT. DREDGE

increased construction cost, but the author is inclined to the opinion that it would reduce operating costs sufficiently to warrant some increase in this direction beyond common practice.

88 **The brakes on the ladder hoist must be adequate in capacity**

and made so that there may be no possible danger of their giving way. If the brakes on the main winch should give way little damage is likely to occur, but if they fail on the ladder winch a broken digging ladder is the probable result, if nothing worse. The following arrangement has been found very satisfactory. Two band brakes are used, one being operated by hand from the pilot house, the fixed end of which instead of being fastened to the deck is attached to a lever arm, which through a cross shaft operates the other brake, thus giving almost unlimited power with very little effort.

89 On most ladder hoists the drum-shaft gear is keyed to the drum shaft and it is nearly always located very close to the drum. It is a much better plan to design the side of the drum in such a way that the gear rim can be bolted to it, thus doing away with the gear spokes, making a much stronger job and taking all driving strain off the drum shaft.

MAIN WINCH

90 The main winch (Fig. 23) operates the swinging lines, both bow and stern, raises the spuds, and is usually fitted with one or two head-line drums for wrecking and repair work and to step up with when one spud has been broken. The winch is driven by a variable speed motor which together with the levers actuating the clutches is operated from the pilot house. Some form of friction clutch is commonly used on each drum. The writer has for some time been using an electro-magnet clutch on the pulley shaft of the winch similar to the one described for the bucket drive, and jaw clutches for each drum. The jaw clutches are extremely simple and require no attention, whereas a friction clutch is likely to get out of adjustment. Also each winchman wants his clutch set differently and unless great vigilance is exercised they will be constantly adjusting these clutches, which means lost time. The friction clutches on each drum are easier to operate, but with a little practice the jaw clutches give perfect satisfaction.

91 A good many dredges have a separate drum mounted on the after part of the dredge driven from the countershaft of the screen drive and is used to raise and lower the stack ladder. The drum is driven by a worm gear of such pitch that it will not run backwards from the weight of the ladder. Other dredges have an extra drum mounted with the main winch for this purpose. It is kept from reversing by a pall and ratchet. Considering the little use this drum gets its location and design seem to be of very little moment, but a worm drive and separate drum mounted aft are preferable.

92 *Swinging Lines.* On deep ground and for the larger dredges the bow swinging lines pass from the drum to a sheave or combination of sheaves located near the bow, then to a sheave on the shore, and from there to the lower end of the ladder. By this method of rigging very little side strain is put on the digging ladder or well hole. The disadvantage of this method is the necessity of a shore sheave and the consequent wear and time lost in moving and handling it. Where the ground is not too deep, a single line, which is simple to operate but which puts a large side-bending strain on the ladder and well hole is used. If the ladder and hull are built strong enough, this is not an objection. For the 13- and 16-ft. dredges a single line would have to be very large and would hardly be advisable.

93 *Brakes.* All winch drums are provided with band brakes operated from the pilot house. The connection between brakes and clutches and pilot house are made through bell cranks, rods, etc.

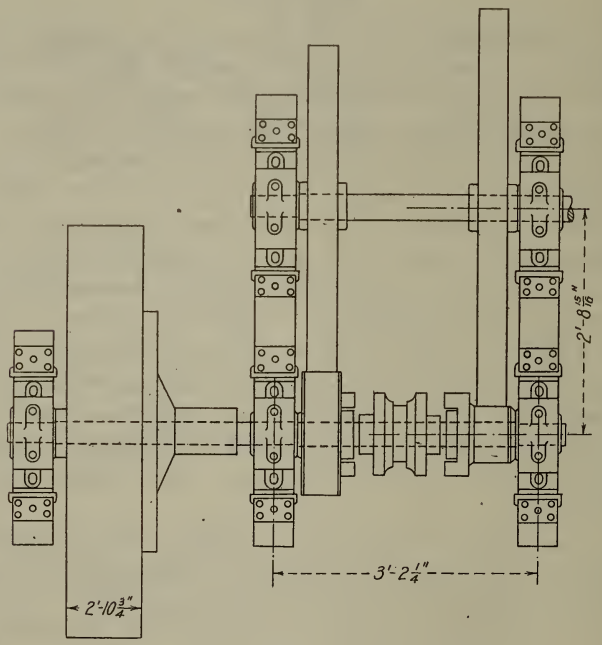


Fig. 23-a Magnetic Clutch and Drive for Main Winch, Fig. 23-b

Flexible wire rope running over sheaves is used in several instances for the brakes.

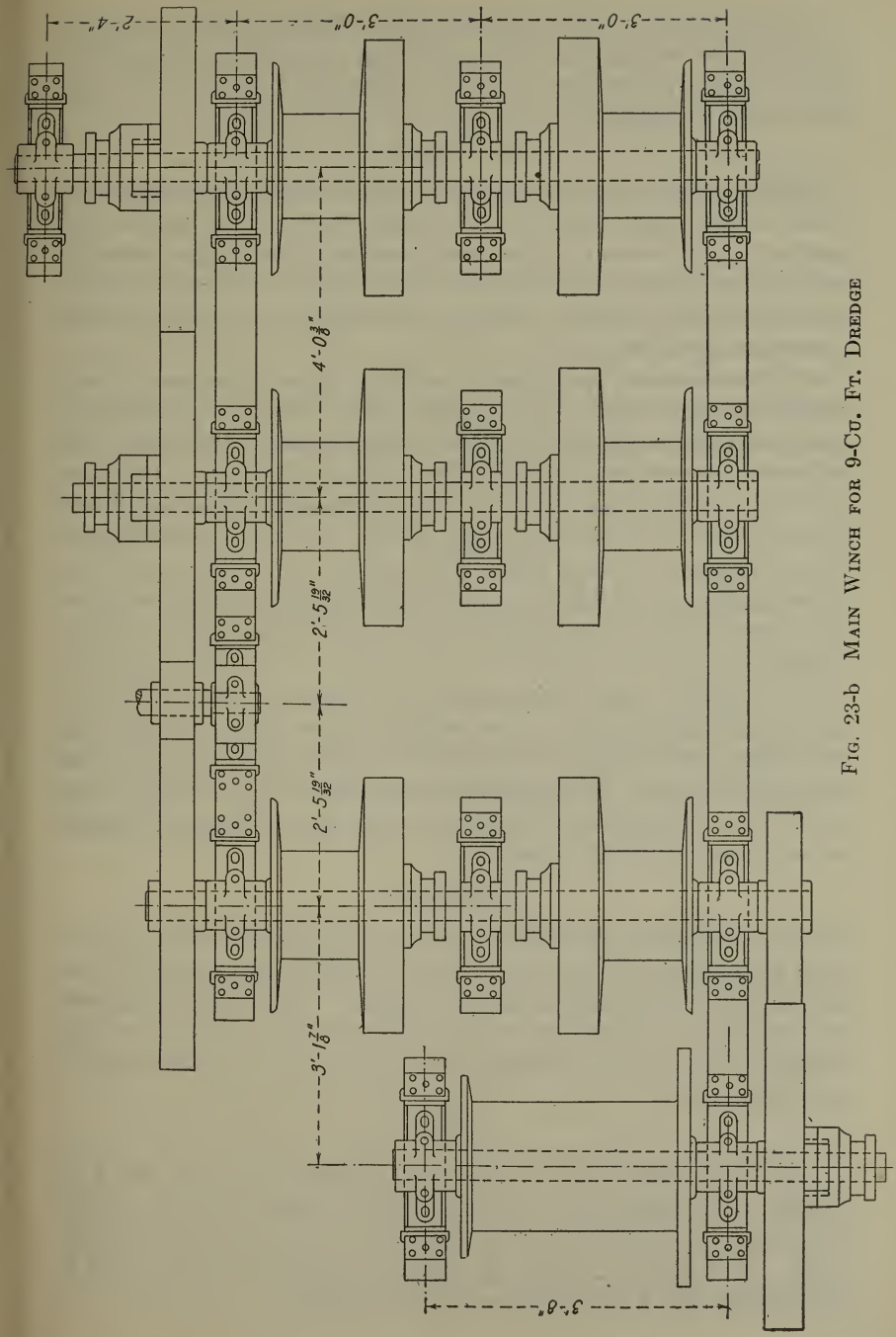


FIG. 23-b MAIN WINCH FOR 9-CU. FT. DREDGE

94 *Foundations.* Cast iron, cast steel, structural steel and wood are used for foundations. The author has been using wood in his recent designs on account of its producing less noise and vibration than metal foundations.

WIRE ROPE

95 Wire rope is used on the dredges for hoisting the digging and stack ladder, spuds, for all the swinging lines and for guying the gantries. On account of the small diameter sheaves, the necessity of the ropes being under water part of the time and the general hard service they get, their life is short as compared with ropes used for most other purposes.

96 Some dredge constructors use the regular open and closed sockets for their guy rope ends, and others think this practice is dangerous and use clips. The writer has never had any trouble with standard rope sockets if properly attached to the rope end. This may be done by first spreading the wire strands, treating them with dilute hydrochloric acid, inserting in the socket with the strands spread broom fashion, and then filling with melted zinc. The ends of the wires should not be turned over, but left straight. If turned over some are likely to get more strain than others, and to break and cause trouble.

97 Different makes and types of wire rope are used. The author's experience points to standard 6/19 plow steel hoisting rope for ladder, spud and side lines and 6/7 crucible cast steel standing rope for the guys. Hot Stockholm tar makes a good dressing, also most of the rope greases to be had on the market. Whatever dressing is employed it should be used freely.

SHEAVES

98 The sheaves used for the various purposes are made of cast steel in preference to cast iron. They are bushed with bronze or cast iron and their hubs made as long as possible. When long hubs cannot well be used, as in multiple blocks, the pins should be made of large diameter, so that the pressure will not be too great on each unit of resisting surface.

PUMPS

99 Three centrifugal pumps are ordinarily used on the gold dredge. The high-pressure pump should be designed to have a working head of at least 60 ft.; more is better if the gravel is at all hard to wash. This pump furnishes the water for the spray pipes or jets in the screens and is direct-connected to its motor.

100 The low-pressure pump is likewise driven by a direct-connected motor and should have a head of 20 to 30 ft. This water is piped to the tables where needed, depending on the nature of the gravel and the arrangement of the gold-saving tables.

101 The priming or fire pump is usually a two-step pump with a working head of 100 ft. or more. This pump is used in case of fire, for washing down decks, in cleaning up, for pumping out the bilge, and for priming the larger pumps.

102 All pumps should be fitted with both foot and check valves Bypass piping equipped with the requisite valves should be put in to be used for priming. The foot valves are provided with a grating, and the suction should be still further protected by a wire netting box attached to the side of the dredge. If this box be made of ample dimensions, say, 8 ft. by 15 ft., a $\frac{3}{8}$-in. mesh will work well. If a smaller box be used, $\frac{1}{2}$- or $\frac{5}{8}$-in. screen will have to be used, otherwise it is liable to get stopped up.

MOTORS

103 Three-phase induction motors are used on all standard California dredges, variable-speed for the digging and winch motors and constant-speed for the other pieces of machinery. Lately a form of variable-speed motor has been used for the stacker and screen drive, which gives a better starting torque for getting these parts under way and is particularly useful when it is necessary to start under full or overload. Any of the standard types of motor give most excellent results if the proper sizes are provided.

TRANSFORMERS

104 It is customary to bring the current onto the dredge at 2200 or 4400 volts, then transform down on the dredge with three oil-cooled transformers to 440 volts. Many of the later boats are using 2200-volt motors, thus avoiding the use of transformers altogether. Beside the three main transformers, another small one reduces the current to 110 volts for lighting purpose. All of the transformers are mounted in steel or asbestos-lined compartments opening to the outside of the dredge. They have pipe connections which allow the oil to be drained off and conveyed overboard to a point some distance from the transformers. These precautions are necessary in order to secure the best insurance rates and are to be recommended as a safeguard against fire. Another precaution which might prove useful is to provide the transformer housings with plate steel doors hung from

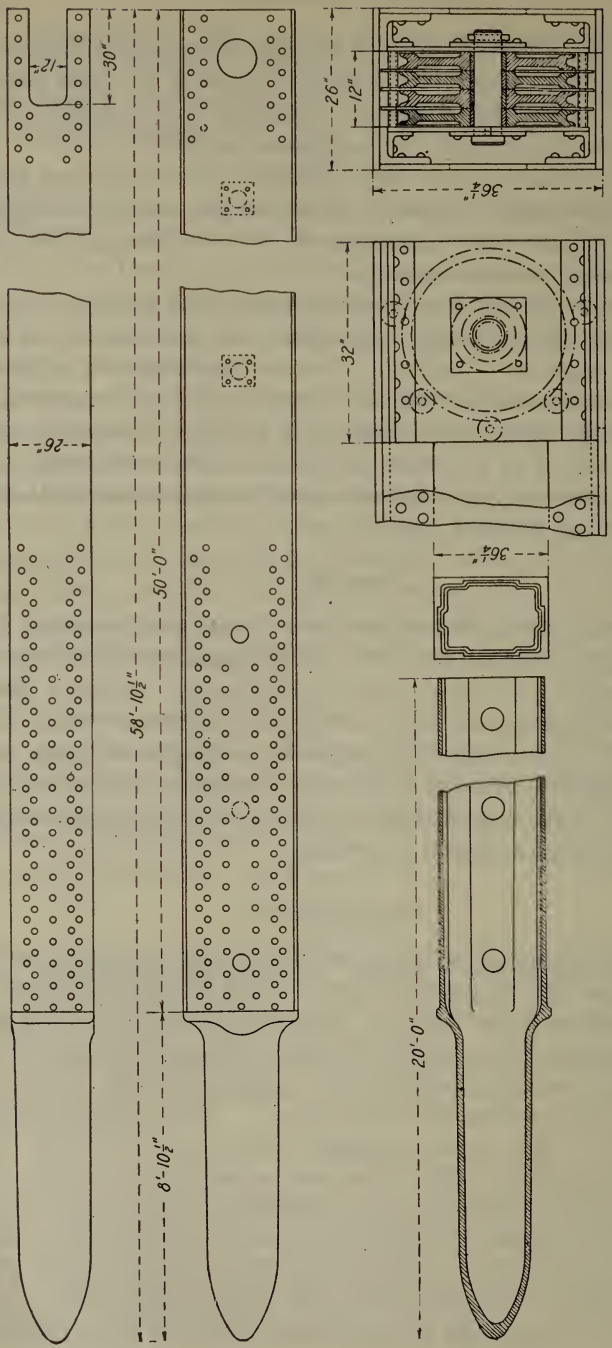


FIG. 24 SPUD FOR 5-CU. FT. DREDGE, SHOWING SECTIONS

the top with an automatic latch at the bottom and so constructed as to be readily dropped and latched in case of fire.

WIRING

105 The electric current is brought on board the dredge on floats in a three-conductor armored cable at a point near the stern, this part of the hull having the least movement. An oil switch is located near this point and is often so arranged that it may be instantly thrown out from the pilot house. From this switch the current is conveyed to the transformers in case low-voltage motors are used and from there the secondary is distributed to the different switchboards. The latest boats use conduits and condulets for all the wiring. If open wiring is used the conductors should not be stretched too tightly and they should all be stranded. The single wires are likely to break under the severe vibration of digging and the movement of the hull and superstructures. Stranded conductors even with the conduit wiring are recommended by the writer. For the motors located on the lower deck their wiring connections can be made through the hold. For the other motors the best method must be determined according to the conditions.

SPUDS

106 The early California dredges used a head line to dig on and even after the spud system was introduced it was a mooted question which was the better. The standard California type dredge uses the spud system (Fig. 24) exclusively, the head line being abandoned except in a few cases where the nature of the gravel or other conditions are unusual. At first, wooden spuds were used, then one wooden and one steel, the wooden spud being used to step up with and the steel one to dig on. Now both spuds are often made of steel. This plan is by far the best, since if one breaks the other may be used to dig on, using the head line to step up with. If only one steel spud is provided and a break occurs, the dredge must be shut down at once and the broken spud taken out, repaired and put in again before any digging can be done. Several forms of spud are used, and their make-up can best be understood by referring to Fig. 24, which shows views and cross-sections of a common design. An extra cover plate riveted to the tension side of that portion of the spud having the greatest strain adds strength with comparatively little increased weight.

107 The lower ends of the spuds are protected by cast-steel points, which extend far enough into the structural part to give ample

riveting area. The later designs are made longer than the older ones (Fig. 22). At the bottom they are acorn-shaped and change gradually from a circular to a rectangular cross-section at the point where they enter the structural portion of the spud, and they often extend 8 or 10 ft. below this point.

108 Heavy car springs are often used in the spud frames to reduce the shocks of digging and those using them think that they lessen the danger of breaking very much. This is doubtless true, but the author is of the opinion that it also reduces the digging capacity of the dredge and that the best remedy for broken spuds is to build them stronger.

CRANES AND REPAIR DEVICES

109 The modern dredges are provided with traveling or boom cranes covering the heavy machinery so that repairs and replacements may be made expeditiously and without calling in more men than the regular dredge crew. A thwartship I-beam with traveler, or a swinging boom covering the forward portion of the dredge and bucket line, is used for changing buckets. Brackets are riveted to the ladder, into which chains are hooked to hold the bucket line after parting. It is well to have an I-beam or at least an eye or hook well secured above the ladder hoist, main winch, pumps, transformers and screen. Over the bucket line a thwartship traveling crane is most useful. It should be of sufficient capacity to lift the upper tumbler, shaft and gears and should extend far enough so that these parts may be lowered over the side of the dredge. If a hatchway is provided through the upper deck on each side of the bucket drive this crane may also prove useful for handling parts on the lower deck.

HULL

110 The hull is built of Douglas fir and is proportioned to accommodate properly the machinery which it supports. Before designing the hull the general dimensions and weights of all machinery should be determined and the hull built accordingly. The depth largely depends on the weight to be supported, the beam on the gold-saving area desired, the length and width of well on the dimensions of the digging ladder, and the digging ladder in turn on the depth of gravel to be dug. The distance from well to stern is determined by the length of screen and the general distribution of machinery fore and aft. The hull rides on an even keel when digging at an average depth and should have a free board of not less than one-fourth its total depth.

111 The cross-section is rectangular, with the deck slightly crowned. Thwartship frames are used, the bottom planking being spiked to the bottom thwartship timbers, the decking to the upper timbers or deck beams, and the side planking bolted to the upright side posts. Besides these, two or more pairs of posts are put in each frame and to these the well planking and bulkheads, or fore-and-aft trusses are bolted. Clamps or fore-and-aft strengthening timbers are bolted inside the frames at the corners where the posts join the deck beams and bottom thwartship timbers. The size of these clamps depends on the size of the hull. Drift bolts are driven in the side planking and bulkheads, each bolt passing through $2\frac{1}{2}$ planks. They may be put in as thickly as desired, and add considerable strength to the planking and bulkheads. Four-inch deck and bottom and 6-in. side planking is commonly used. The stern is often planked with the same size as the sides, put edgeways instead of flat. At the point where the spuds are located a short bulkhead or extra bracing is often used. Just aft of the well and below the bucket drive, one or more vertical cross trusses are often used, likewise horizontal cross bracing is put in on top of the thwartship frame timbers along the compartments adjacent to the well and extending some distance aft from the well throat.

112 The bow is made narrower than the rest of the barge, the frames being shortened and the side planking steamed and bent around them. How much this portion of the hull should be narrowed down will depend on the width of cut it is desired to make, together with the distance the digging ladder projects beyond the bow. Using one spud as a center and drawing the outline of the hull, digging ladder and cut to scale, will graphically illustrate how much to bring in the bows so that the dredge may dig its way along any desired width of cut.

113 The above is a general description of the most common type of hull construction. The writer has been using what he considers a somewhat stronger construction, but one which is more expensive. Fore-and-aft stringers are laid below the thwartship timbers. The posts are bolted to both the fore-and-aft and the thwartship timbers, the bottom planking being laid athwartships. Clamps are used above the thwartship timbers the same as in the other form of construction, the difference being that there is an extra set of fore-and-aft timbers, to which the bottom planking is spiked, this planking running athwartships instead of fore and aft. This construction requires more lumber and labor, but is stronger, and by using thwart-

ship planking full lengths can be used, doing away with all butt joints. The tendency of dredge hulls is to sag along the fore-and-aft center line. Where fore-and-aft planking is used this movement of the hull tends to open up the seams, but where thwartship planking is used the strain is lengthways of the planking. When a hull is new these weaknesses do not show, but after years of service all such little things count, and since the life of a hull determines the life of a dredge, it pays in the long run to use every precaution possible to add to the stability and length of life of the hull. Solid fore-and-aft bulkheads are recommended, also the use of well plates made up of 6 in. by 14 in. planking (for the 7 to 9 cu. ft. dredge), laid horizontally below the deck beams and filling the roof of the first compartment on each side of the well for 30 ft. each way fore and aft from the throat of the well. These well plates are particularly useful in dredges having long well holes.

114 Another plan which will add strength at small cost is to put in diagonal wooden braces from the fore-and-aft timbers near the bottom of the intermediate bulkhead to the clamp near the top of the well hole. This will aid materially in preventing sag toward the well hole.

115 Still another form of strengthening construction being used, particularly on the larger boats, is an overhead thwartship truss with heavy hog rods running diagonally from the upper chord near the sides of the dredge to the bottom of the hull at the well, thus preventing sag at this point.

116 The bottom, sides, bow, stern and deck are all calked with oakum and the seams pitched or similarly treated. The inside is treated with carbolinium, Stockholm tar, painted or left without any treatment whatever. On the outside below the water line the planking may be painted or not as desired. Above the water line all timber work is thoroughly painted. Before putting timbers together all joints are painted with white lead, mineral paint, or coated with tar or carbolinium.

117 The deck beams are extended beyond the sides of the barge at the after portion of the dredge. Braces are put under these in order to support the housing uprights. This gives extra room for gold-saving area or passageways. Hatchways are provided for entrance into each compartment, also a system of ventilation should be installed, using either mechanically driven fans or natural ventilation induced by cowled ventilators. Proper ventilation will do much towards prolonging the life of the hull. Sea cocks are often installed

in order to sink the dredge in case of a fire which has gone beyond hope of being extinguished. If sea cocks are used they should be located near the bows where a hot fire in the housing will not prevent their being approached and quickly opened.

118 *Main Truss.* A large proportion of the California dredges have a truss along the sides of the well and extending from bow to stern gantry, with wooden chords, posts and braces and steel hog rods. This truss is made 15 to 25 ft. high, the upper chord often being used to support the upper tumbler bearings and the upper end of the digging ladder. In the earlier dredges there was a tendency for the hulls to go down at bow and stern and hump up near the throat of the well. This truss assists in preventing this condition, strengthens all the gantries and spud frame, and makes an excellent support for the housing, screens, etc. It should be designed as a truss supported at the middle with weights suspended at each end.

119 *Bow Gantry.* At the bow is a gantry (Fig. 25) which supports the forward end of the digging ladder, the details varying in the different dredges. A steel cap of box or double I-beam section on four posts is a common type. In some dredges the four posts are parallel; others have two posts parallel to the well sides and two sloping inward joining the first at the cap. Still others have the four posts parallel for half their length, the two outside ones being cut at this point, and from here diagonal braces run to the intersection of cap and inside posts. The whole gantry is inclined forward in order that it may have a more direct pull on the digging ladder. In a good many of the California dredges the gantry posts go through the deck and are stepped on heavy foundation timbers or castings near the bottom of the hull. A few dredges have rigid foundations for these posts on the deck, and at least one uses a socket casting into which the posts fit loosely.

120 This gantry has thwartship bracing and is guyed by ropes or tie rods to the main gantry, hull timbers or both. The top chord of the main truss runs to the middle portion of this gantry and is fastened to it by means of steel plates. It is also customary to run a brace diagonally from near the cap to the top of the first truss post. Some form of pin and plate clamp is provided for the bow gantry cap, to which is attached the ladder hoist blocks and gantry guys.

121 *Main Gantry.* The main gantry is a support for the bucket drive and the after end of the digging ladder. It has a very severe duty to perform and should be built in a most substantial manner. The author prefers to place a post under each bearing and use heavy

lattice bracing between the posts. Too many plates should not be used to tie the timbers together, and arrange matters if possible so that all bolts will be accessible. Where many plates are used it is very difficult to assemble the parts and in case of repairs the delay

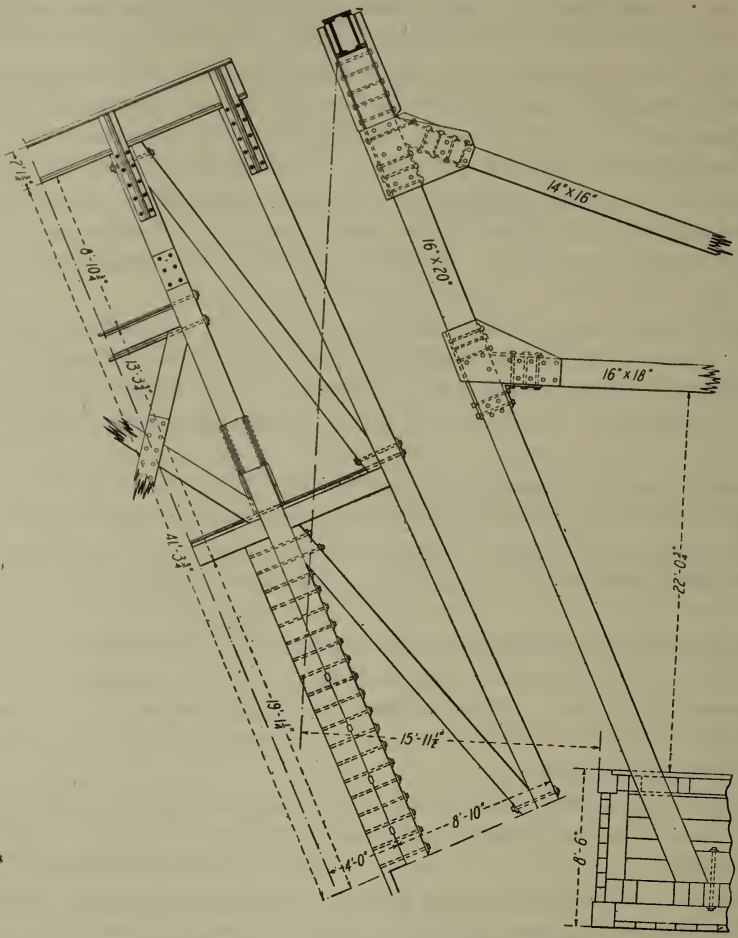


Fig. 25 Bow Gantry, 9-Cu. Ft. Dredge

is great in taking apart and re-assembling, owing to the difficulty of lining up the holes in the different plates.

122 *Stern Gantry.* The stern gantry (Fig. 26) supports the spuds and the stack ladder and must be made high enough to raise the

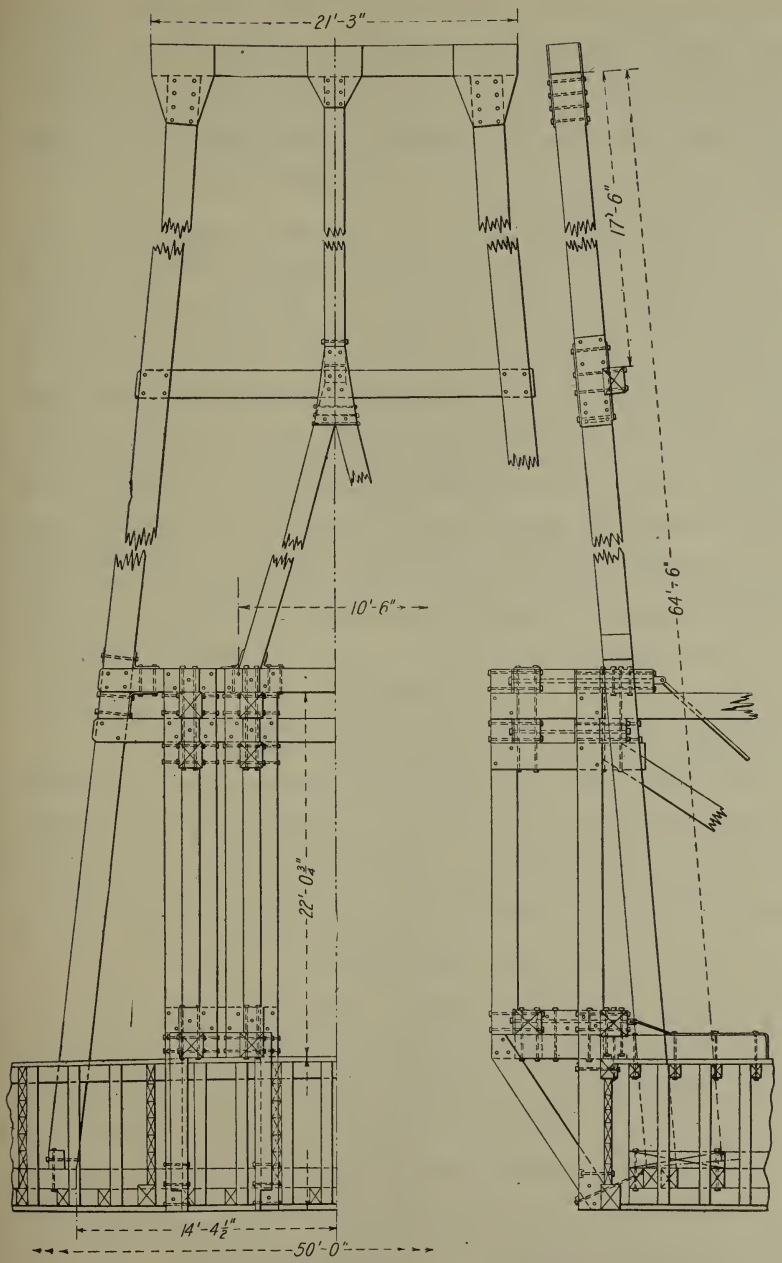


FIG. 26 STERN GANTRY, 9-CU. FT. DREDGE

spuds so that they will nearly clear the bottom of the hull. It is inclined slightly aft so that the spud hoist blocks will be vertically above the center of the spuds. The design is simple, two, three or four posts being used with a wooden or steel cap. Clamps are used for the spud hoist blocks, ladder support and guy ropes. The posts may rest on foundation timbers or castings located near the bottom of the hull or on the deck.

123 *Spud Housing.* At the stern of the dredge the two spuds are enclosed in a housing which allows them to slide up and down but holds them in a vertical position. At some point above the deck, usually at the elevation of the upper chord of the main truss, heavy timbers are framed around the spud, with tie rod connections to the hull or main gantry. This framing together with the tie-rod connections resists the tendency of the upper part of the spud to move backward as the buckets engage the gravel and thrust the hull toward the rear. This clamp must be made strong enough to withstand the full digging power of the dredge, increased to some extent by the slight fore-and-aft movement of the hull and the consequent momentum of the full weight of hull and machinery. The writer does not approve of running the tie rods to the main gantry, since no matter how strong this part is made the constant pulling and jerking on one side will tend after a time to throw the bucket-line shafting out of line. The guy rods should be run into the hold or to some other part which does not support shafting.

124 Many different spud-housing designs are used, but the present tendency seems to be to use heavy castings covering a large area of the stern to resist the forward pressure of the spud and a structural or cast-steel clamp 20 ft. or so above the deck. In order to keep the spud from moving backward heavy steel keepers are used which can quickly be removed when it is necessary to change spuds. The timber construction will work well enough, but it is likely to get loose in time, and it is difficult to design a wooden structure so that it will be at the same time strong and easily removable. Whatever form of spud housing is used it should be so enclosed that small rocks can not lodge where they will be caught between the spud and its guide.

125 *Housing.* The after portion of the dredge is housed in, this housing covering the gold-saving tables, pumps, main winch, ladder hoist and transformers. The after central portion is extended up and covers the screen and screen drive. A covering should also be provided for the bucket drive, but this is not essential. The roof of the housing is made with a slight crown or slope to the sides and

covered with painted canvas, ship fashion. The roof joists should be made heavy enough to support a considerable weight, since in repairing it is often convenient to lay down one of the big gears or the like, and for this reason the flooring should be at least $1\frac{1}{4}$ in. thick, or made of two thicknesses of planking. The stern and sides along the gold-saving tables are not usually boarded up. It is thought that an unobstructed view lessens the danger of amalgam stealing.

126 *Pilot House.* The pilot house from which the digging motor and winches are handled is usually located on the starboard side of the upper deck above the main winch and a little forward of the bucket drive. This is the common location, but the pilot house on the central line of the dredge above the bucket line and just forward of the bucket drive seems preferable. The only objection to this location is that the lever connections to the winches are a little more complicated. This location gives an excellent view in all directions around the dredge and down the bucket line. By proper arrangement of gratings the main winch may be seen. The bucket drive, stacker and spuds are also within sight of the winchman. In the writer's opinion there is no question as to the proper location for the pilot house, and all his dredge designs have their pilot houses well up in the center of the boat.

DIMENSIONS

127 In Table 1 the more important dimensions and some other data of several California gold dredges built at different times, of different sizes, and for different conditions are given. Most of the data were kindly furnished by the makers, and will serve to give a general idea of typical dimensions. In checking over the figures furnished several discrepancies were noted, but on the whole the figures are no doubt approximately correct.

CONSTRUCTION COSTS

128 The construction costs vary greatly, of course, according to the quality of workmanship and material desired, facilities of the builders in the way of properly equipped shops and foundries, and the geographical situation of the ground on which the dredge is to be built. The following may be considered a fair average of many cost records in California, for rough estimating purposes:

129 For all the machinery, including motors, pumps, wire rope, electric supplies, etc., $12\frac{1}{2}$ cents per lb., delivered on the ground; for installing, $4\frac{1}{2}$ cents; or a total of 17 cents per lb. for the machinery installed. The lumber costs about $30 per 1000 ft. at Portland, and

this together with freight, bolts, nails, hog rods, steel plates, oakum, paint, etc., will make the hull material cost about $50 per 1000 ft. of lumber used; labor of building costs about $40 per 1000 ft., giving $90 per 1000 ft. as a total cost of the completed hull. In addition the direct costs chargeable to hull or machinery, insurance, traveling, superintendence, camp equipment, temporary buildings, shop, derrick, office expense, design, etc., amount to between $10,000 and $25,000, depending on whether new designs have to be gotten out, what construction tools and equipment are on hand, the size of the dredge, and location of the ground on which it is to be built. For a a complete dredge, $180 per ton, on a basis of its displacement, is a fair average.

TABLE 1 GENERAL DATA ON CALIFORNIA GOLD DREDGES

Name of maker	Risdon	Bucyrus	Link Belt	Risdon	Marion	Marysville Dred'g Co.
Date built	September 1897	1907		1910	1905	1911
Capacity bucket, cu.ft.	3.2	5	5	5.6	7.5	9
Depth to dig below water line, ft.	20	38	35	30	65	65
Depth to dig above water line, ft.	8	10	12	10	10	10
Nature of digging	Loose	Medium hard	Hard	Medium	Medium	Medium
Hull:						
Displacement with all machinery	3 ft. 6 in.	5 ft. 9 in.	5 ft. 8 in.	5 ft. 3 in.	6 ft. 6 in.	6 ft. 2 in.
Length, ft.	70	102	106½	88	115	115
Depth, ft.	5	7 ft. 9 in.	7 ft. 10½ in.	7 ft. 6 in.	8	8½
Width, ft.	29	36	40	36	40	50
Overhang on each side, ft.	0	2	....	0	2½	4
Length of well, ft.	42	56 (approx.)	51 ft. 1 in.	48	63	63
Free board, ft.	1½ (approx.)	2 ft. 3 in.	2 ft. 2½ in.	2 ft. 3 in.	1½	2 ft. 4 in.
Side planking, ft.	3	4	4	4	4	6
Bottom planking, ft.	3	4	4	4	4	4
Height of bow gantry above deck, ft.	22	36	34 ft. 6 in.	34	39 ft. 6½ in.	37
Height of stern gantry above deck, ft.	24	50	61 ft. 8 in.	36	58 ft. 9 in.	58
Height of main gantry above deck, ft.	16	23	21 ft. 8 in.	21 ft. 6 in.	20 ft. 2 in.	22
Total amount of lumber in hull, gantries, housing, etc., ft.	75,000	136,000	263,600	156,000	300,000	400,000
Digging ladder:						
Weight, lb.	14,000 (approx.)	48,400 (approx.)	53,575	33,000	79,000	94,400
Distance between centers, ft.	40	87	84½	71 ft. 6 in.	114 ft. 1 in.	115
Extension beyond bow, ft.	12 (approx.)	25½	26	18	38	43
Width, ft.	3	4½	5 ft. 8 in.	3 ft. 8 in.	4 ft. 7½ in.	4 ft. 7½ in.
Maximum depth girder, ft.	24	7	5 ft. 6 in.	54	....	8
How upper end is supported	Fixed hanger - shaft with telescope bearing	Concentric	Steel pin, 8 in.	Fixed hanger - shaft with telescope bearing	Concentric	Concentric
General design	Plate girder	Plate girder	Plate girder	Plate girder	Lattice girder	Lattice girder
Stacker:	Bucket elevator	Belt conveyor	Belt conveyor	Bucket elevator	Belt conveyor	Belt conveyor
Length ladder, ft.	42	90	106	72	136	139
Width ladder, ft.	3 ft. 2 in.	4	3 ft. 8¾ in.	3 ft. 8 in.	3 ft. 8 in.	4 ft. 4 in.
Depth ladder, in.	15	5	4	15	8	7
Weight ladder, lb.	12,000	18,000	15,000	19,365	22,500	24,000

TABLE 1 GENERAL DATA ON CALIFORNIA GOLD DREDGES—*Continued*

Name of maker.....	Risdon	Bucyrus	Link Belt	Risdon	Marion	Marysville Dred'g Co.
General design.....	2 I-beams	Lattice girder	Lattice girder	2 I-beams	Lattice girder	Lattice girder
Width belt, in.....	Buckets 15 in.pitch	30	30	Buckets 15 in.pitch	32	32
Speed belt.....	48 buckets per min.	350 ft. per min.	Variable	54 buckets per min.	350 ft. per min.	400 ft. per min.
Horsepower motor...	Driven by main engine	15	20	10	25	35
Maximum height of discharge, ft.....	20	30	35	30	50	54
Bucket line:	Open connected	Close connected	Close connected	Open connected	Close connected	Close connected
Number in line.....	30	82	93	40	93	84
Maximum speed of buckets per min...	16	22	22	14	19	20
Weight bucket complete, base, hood, lip, pin, bushing, lb.	570	1300	1490	1200	2187	2268
Pitch of bucket, in...	24	28 5/16	24½	25	32¼	36
Number front eyes..	2	2	2	2	3	2
Number back eyes..	2	1	1	2	2	1
Length back eye, in..	4	16	19⅜	6	8⅛	16¾
Width front eye, in..	4	3⅝	6½	6	3¾	4
Diameter pin, in.....	1⅞	4⅛	4½	2¾	4⅝	5¼
Length pin, in.....	10½	23¼	30⅞	12½	2 ft. 3 5/16 in.	24⅞
Ladder roller:						
Number on ladder...	6	12	19	10	17	17
Diameter in.....	8	14	12	12	14	14
Diameter, shaft in roller, in.....	2	3 15/16	3 7/16	3½	4½	5 7/16
Diameter shaft in bearing, in.....	2	3⅜	3 7/16	3½	3 15/16	5
Length shaft in bearing. in.....	6	11⅛	....	8	12	11⅞
Material used.....	White iron	High carbon and manganese steel	Semi steel	Manganese steel	High-carbon steel	Chrome steel
Weight roller and shaft, lb.....	280	1030	1008	778	1160	1140
Upper tumbler:						
Diameter shaft in tumbler, in.....	8	14	14	11	17	17
Diameter shaft in bearing, in.....	8	12	12	11	14	14
Span between bearing...	4 ft. 6 in.	6 ft. 6 in.	....	5 ft. 2 in.	9 ft. 3 3/16 in.	9 ft. 3 3/16 in.
Weight complete with shaft and plates, lb.....	3010	11,200	10,185	6,220	18,750	21,310
Lower tumbler:						
Diameter shaft in tumbler, in.....	6	10¼	11¾	10	11¼	12
Diameter shaft in bearings, in.....	6	9	7¾	10	10	10

TABLE 1 GENERAL DATA ON CALIFORNIA GOLD DREDGES—*Continued*

Name of maker.....	Risdon	Bucyrus	Link Belt	Risdon	Marion	Marysville Dred'g Co.
Length of shaft, ...	4 ft. 2 in.	4 ft. 9¼ in.	6 ft. 5⅞ in.	5 ft. 3 in.	4 ft.11¾ in.	5 ft. 3½ in.
Maximum height side flanges at corners above wearing plates, in.....	7¼	6⅝	9½	11	12½	11¼
Thickness side flanges at top and bottom, in.....	1-2¼	1⅝-3⅛	2¼-3	1½-3¼	1-1⅜	2
Inside flare of side flanges	....	2 in. in 12 in.	Curved	....	2⅜ in. in 12 in.	Straight
Weight complete with shaft and plates, lb.....	4,176	10,000	8,355	7,746	11,340	12,758
Ladder hoist:						
Total weight, lb.....	Part of	14,800	20,000	5,660	....	39,000
Diameter of drum,in.	main winch	18	30	18	20	3 ft. 6 in.
Diameter of drum shaft, in		6½	6 7/16	....	8	9½
Speed of drum, r.p.m.		19.2	9.6	....	12.8	7.6
Maximum speed ladder line, ft. per min.	....	....	....	10	67	100
Horsepower required to raise ladder at above speed.....	....	....	....	35	....	120
Diameter of sheaves, in.....	....	30	30	36	30	42
Main winch:						
Total weight, lb.....	10,385	23,700	32,500	24,200	....	23,275
Horsepower of motor	12	20	35	25	25	25
Diameter of drums, in....	15	16	18	18 and 24	18	16
Diameter of drum-shaft, in.....	3½	4⅞	4 11/16	5⅛ and 7	4 11/16	4 11/16
Maximum number r.p.m. drum.....	....	14	20.5	2	9	16
Minimum number r.p.m. drum.....	....	2.6	....	....	2.5	4.7
Maximum speed of side swing at end of digging ladder, ft.....	....	62	....	....	17.5	38
Minimum speed of side swing at end of digging ladder. ft.....	....	11.5	....	....	5	11
Speed of raising spud, ft.....	....	10	....	....	4.5	9.5
Horsepower necessary to raise spud at above speed....	....	10	....	....	6	15

TABLE 1 GENERAL DATA ON CALIFORNIA GOLD DREDGES—*Continued*

Name of maker	Risdon	Bucyrus	Link Belt	Risdon	Marion	Marysville Dred'g Co
Bucket drive:						
Horsepower digging motor	50 h.p. steam	100	100	100	150	200
Diameter and face engine pulley, in.	66	96 x 24	86 x 21	96	96 x 24	96 x 25½
Pitch diameter and face pulley shaft pin	....	10.82 in. P.D. x 7 in. F.	11.52 in. P.D. x 6½ in. F.	....	11.936 in. 8 in. F.	11 936 in.–8¼ in. F.
Speed pulley shaft, r.p.m.	....	160	143	156	113	115
Speed intermediate shaft, r.p.m.	....	44	22.5	56	21	21
Pitch diameter and face intermediate shaft gears	....	39.67 in. x 7 in.	74.49 in. x 6 in.	47¾ in. x 6 in.	62.87 in.–8 in. F.	62.87 in.–10 in. F.
Pitch diameter and face intermediate shaft pinions	....	13.5 in. x 8 in.	14.32 in. x 10 in.	....	20.136 in.–10 in. F.	20.136 in. –10 in. F.
Speed tumbler shaft, r.p.m.	8	4½	4.5	7	3.1	3
Height tumbler shaft above deck	....	23 ft.	25 ft. 2½in.	....	22 ft.5¼ in.	24 ft. 6 in.
Pitch diameter and face tumbler shaft gears (bull wheels)	....	9 ft. 11 in.–8 in. F.	71.62 in.–9 in. F.	....	143.55 in.–10 in. F.	143.55 in.–10 in. F.
Weight bull wheels, lb.	2,100	5,500	3,110	3,200	8,985	10,200
Spuds:						
Total weight including all castings, lb.	....	20,400	43,800	....	32,000	36,000
Total length, ft.	....	50	58 ft. 10 in.	....	60	61
Width, in.	....	24	26	....	24	24
Depth, in.	....	36	36	....	36	36⅞
Type of construction	....	Box type	Box girder	....	Center web	Center web
Screen:						
Type	Revolving	Revolving	Shaking	Revolving	Revolving	Revolving
Total length, ft.	20	25½	32	24	30	36 ft. 3 in.
Width or diameter	3 ft. 6 in. diameter	6 ft.	7 ft. 4 in. upper 7 ft. 11 in. lower	4 ft. 6 in. diameter	6 ft. diameter	6 ft. diameter
Size of holes	⅜ in. dia.	⅜in.to½in.	⅜ in.	⅛ in. dia.	½in.to⅝in.	½ in.
Grade	1½ in. in 12 in.	2 in. in 12 in.	1⅞ in. in 12 in.	1½ in. in 12 in.	2 in. in 12 in.	2 in. in 12 in.
Horsepower of motor	Driven by main engine	20	20	15	35	50

TABLE 1 GENERAL DATA ON CALIFORNIA GOLD DREDGES—Continued

Name of maker.....	Risdon	Bucyrus	Link Belt	Risdon	Marion	Marysville Dred'g Co.
Number r.p.m. or strokes per min..	12	7	....	9	7	8
Gold-saving tables:						
Total area, sq. ft....	160	800	....	370	1600	2400
Grade.....	1¾ in. in 12 in.	1 7/16 in. in 12 in.	1¼ in. in 12 in.	1¾ in. in 12 in.	1⅜ in. in 12 in.	1⅜ in. in 12 in.
Wood or steel.....	Steel	Wood	Steel	Steel	Wood	Steel
Width of bays, in....	28	36	30	32	33	36
Dimension of riffles..	Cocoa matting and expanded metal	1 5/16 in. x 1¼ in.wood	1 in. x 1 in. x 5/16 in. L.	Cocoa matting and expanded metal	1¼in. x 1¼ x 5/16 in. L.	1¼in. x 1¼ x 5/16 in. L.
High-pressure pump:						
Horsepower of motor	....	15	100	30	50	100
Head, ft.....	18	65	70	31	50	68
Quantity of water, gal. per min.....	3000	500	2800	2400	....	3200
Size suction pipe, in..	....	4	12	14	10	12
Size discharge pipe. in.....	....		10	9	8	10
Make of pump.....	Risdon	Yuba Con. Co.	Worthington	Risdon	Worthington	Abner Doble
Low-pressure pump:						
Horsepower of motor	No low-pressure pump	50	No low-pressure pump	....	25	35
Head, ft.....		60		18.6	25	25
Quantity of water, gal. per min.....	....	2000	....	2200	....	3800
Size suction pipe, in..	....	8	....	....	10	12
Size discharge pipe, in.....	....		....	9	8	12
Make of pump.....	....	Yuba Con. Co.	....	Risdon	Worthington	Abner Doble
Fire pump:						
Horsepower of motor	....	Use high pressure	5	5	10	10
Head, ft.....	....		65	40	90	100
Quantity of water, gal. per min.....	....		150	....	....	....
Size suction pipe, in..	....		3	3	4	2½
Size discharge pipe, in.....	....		2½	2½	3	2
Make of pump.....	....		Worthington	Krogh.	Worthington	Abner Doble
Lines:						
Size ladder hoist, in..	⅝ in.	⅞ to 1	1	⅞ in. diameter	1	1¼
Size stacker hoist, in.		¾	¾	1½in. diameter	¾	¾
Size spud hoist, in...	....	¾	¾	....	⅞ and ¾	¾
Size swinging lines, in....	½—in. rope 300 ft.	¾ to ⅞	¾ to ⅞	⅞ and ¾	1 and ¾	1
Total weight machinery, tons.....	90	260	305	220	300	400

DISCUSSION [1]

R. H. Postlethwaite[2] (written). As the designer and builder of the dredge described as the first successful California dredge, the writer has naturally read the paper with a good deal of interest and wishes to touch upon two points: the bucket tailings elevator or stacker and the gold saving tables, both incorporated in the original design of 1897 and both used at the present day.

Without doubt the most popular stacker is the belt conveyor and for very large dredges operating on a spud, where it is necessary to get the tailings very far back, it is probably the best; but for medium sized dredges operating on a head line, the writer believes the bucket elevator is more economical in annual upkeep, and certainly it is more readily repaired.

The following statistics are from records carefully kept by the late Karl Krug, manager of the New York Machine Shop, who was operating two dredges, Lava Beds No. 2, equipped with a belt conveyor, and Marigold, with a bucket elevator:

	Lava Beds No. 2	Marigold
Total time, months	40	29
Material dredged, cu. yd	2,690,307	1,296,718
Average per month, cu. yd	67,257.67	44,921.31
Total cost of stacking	$6,953.25	$1,812.75
Average cost per month	$173.83	$62.50
Cost of stacking per cu. yd	$0.26	$0.14
Total lost time caused by stacking	526 hr. 27 min.	206 hr. 47 min.
Average lost time per month	14 hr. 3 min.	7 hr. 7 min.

Note: Estimated that only half the gravel dredged goes up the stacker, therefore actual cost per cubic yard stacked is in each case double that given in the table.

These results are from the Oroville field. When a dredge is to be operated in a country far from the base of supplies, the bucket elevator has a great advantage, as any blacksmith or dredge hand can readily repair the essential parts, whereas, a slight accident to a belt conveyor will in a few revolutions put the belt out of commission and it is almost impossible to repair it satisfactorily.

Various features contributing to convenience of operation and durability are introduced in the design of the stacker, such as bushings for the roller shafts, which are hexagonal and can be turned at least three times before being worn out. For lubrication, simple and efficient grease plugs are used.

[1] Given at San Francisco and New York Meetings.

[2] Union Iron Works Co., San Francisco, Cal.

Gold Saving Tables. It is true that mercury riffles are now generally used on California dredges instead of cocoa mats; the fundamental reason being that they do not require such careful attention and in the case of a large company operating several dredges only one reliable gold saver need be employed, the tables being allowed to run for days under lock and key and a general clean-up being made under his supervision every two weeks, during which time the whole dredge must be shut down; whereas cocoa mats require a certain amount of daily attention which, however, can readily be given by the dredge master if he is reliable and competent, and the dredge need not be shut down for the clean-up. A closer actual saving can be made by cocoa mat tables properly operated, especially if the gold is at all rusty, and the following record of tests, made by the Feather River Exploration Company on one of their dredges, bears this out. The experiments were made on a Risdon type of dredge, the mats and riffles alternating from one side of boat to the other:

Experiment	Boat	Duration of test, days	Percentage of total gold saved with equal material treated	
			Mats	Riffles
1	No. 5	32	55	45
2	No. 4	26	51	49
3	No. 4	15	54	46
4	No. 4	15	51.5	48.5
5	No. 4	15	52	48

Note: The experiments were changed from dredge No. 5 to dredge No. 4 for the reason that soon after starting the test the dredge ran into gold that was difficult to amalgamate, while on No. 4 the reverse condition prevailed.

The area of mats and riffles was equal, being 142 sq. ft. each.

J. W. Plant. In designing a gold mining dredge, it is necessary to prepare a strain sheet before the principal parts can be decided upon, for which the following data will be required:

a The pitch and approximate weight of buckets, determined by partially designing the buckets and tumblers, or by choosing the dimensions and weights from some existing dredge. The pitch of the buckets must be sufficiently great so that, when using a six-sided tumbler, there will be space for attaching cushion plates properly after the hub of the tumbler and diameter of the shaft are allowed for. Pitches of 28 in. for 5-ft. buckets and of 33 in. for 7½ to 8½-ft. buckets are none too great.

b The length, approximate weight and upper point of

suspension of the digging ladder. The length is determined by assuming that the ladder will not be depressed more than 45 deg. from the horizontal when digging the maximum depth below the water line; and that when in its upper position, 10 deg. below a horizontal line through the upper point of suspension, the ladder will project far enough beyond the hull so that the buckets will cut sufficient clearance to allow the dredge to be operated without two settings of the spuds.

c The height of the front gantry and the location of the front gantry cap is decided upon with respect to the bow. It is well to locate the center of the front gantry cap as far forward with respect to bow and as high from the deck as can be reasonably done, for by so doing the tension in the ladder suspension tackle is reduced when the ladder is in upper or 10-deg. position.

With these points determined, it is possible to lay out the bucket line whose under side describes a catenary. The writer has usually used a flexible chain of small size to approximate this curve by standing the drawing board on its side and suspending the chain from pins at the upper and lower tumblers. The pitch of the chain is then stepped off carefully along the catenary until an even number of spaces is obtained, which can usually be done in several trials. The catenary must not be too flat or the tension will be excessive.

This operation is gone through for both the 10-deg. and the 45-deg. positions of digging ladder. The tensions in upper and lower ends of catenary are graphically obtained by drawing lines tangent to the circles passing through the centers of pins on the lower and upper tumblers at the points where the catenary comes in contact with them. When these two tangent lines intersect, the combined weight of buckets forming the catenary is assumed to act. This must be true, for if these forces in the same plane balance each other their lines of action must meet in a point.

The upper portion of bucket line, each bucket assumed to be loaded with gravel is considered as having its combined weight located at the center of gravity of the digging ladder. By taking moments about the upper point of suspension of the digging ladder, the tension in the guys and ladder suspension tackle can be determined.

The speed of the bucket line of most modern dredges is taken at 50 ft. per min. The digging motor is usually 75 h.p. for a 5-ft. dredge, 150 h.p. for a 7½-ft. dredge, and up to 300 h.p. for a 16-ft. dredge. Assuming 50 per cent overload on digging motor and neglecting friction of gearing, it is easy to determine the tension in bucket line when its speed per minute is known. This tension must be added to the tension caused by the catenary plus the component due to the weight of the gravel in the buckets on the upper side of the digging ladder in order to obtain the maximum.

Assuming the force acting in the bucket line to act entirely on the bucket coming directly under the lower tumbler and parallel with the deck gives the reaction which must be resisted at the point of the spud. Assuming that the spud fulcrums on the stern of the dredge, gives an approximate means for determining the bending moment set up in this part. When the dredge swings back and forth in making its cuts, a twisting moment is set up in the spud in addition, which can only be approximated. In addition to the above, the dredge has a surging action when digging which sets up additional stresses in the spud which, however, can be made less pronounced by the use of large spiral springs at the upper end of the spud casing. The writer is strongly in favor of this type of construction.

The spuds are usually rectangular in section, built up of plates and angles. Were it possible to make the spud circular in cross-section, it would be preferable, as the twisting moment would not be so pronounced when the spud is imbedded in the sand and the dredge swinging. The bending moments set up in the upper ladder suspension can be determined graphically knowing the tension in the catenary, the weight of the buckets, digging ladder, etc.

The upper tumbler shaft has a very hard service to perform. It is revolving constantly and as the hardness of the digging varies, it is subjected to reversing bending stresses, shock and twisting moments due to the power it transmits. It must be made of material having a high elastic limit and ultimate strength and test bars must show good elongation. Three per cent steel is usually chosen.

The sizes of the various motors having been determined from experience or from existing designs, it is then possible to determine the tension in the ladder hoist cables, spud hoists, swinging

lines, etc. Plow steel is usually used for the ladder hoist cable, and varies from 1⅛-in. to 1½-in. or larger, on 7½-ft. to 16-ft. dredges. The weight to be lifted in the larger dredges is extremely great and to keep from having an excessive number of sheaves, the diameter of the cable must be reasonably large and the quality of cable must be of the best. The diameter of the ladder hoist sheaves on most dredges is smaller than would be tolerated for transmission purposes, being in no case more than 30 times the diameter of cable and in many cases less.

It can be readily seen that if the diameter of the cables is made very large to keep down the number of sheaves, the life of the cable due to bending stresses set up will be very seriously shortened. With two running ends of cable passing into the ladder hoist winch, most modern dredges from 7½ ft. up use 20 or more parts of cable in the tackle. The cumulative stress in the cable due to friction of the sheaves is of course greater in the ends running into the winch than in those parts coming at the bucket end. A factor of safety of five based on published tables of the manufacturers is none too much.

The links between each of the sheaves for supporting them should be packed like the eye bars of a bridge so that the bending moments in the pins will tend to cancel each other instead of being cumulative and thereby keep the diameter within reasonable limits. Sheaves have been made of carbon steel, cast iron and, in some recent cases, of manganese steel. This latter material, due to its great wearing qualities, is suited better than the other materials for this service. Recent tests have shown that it is somewhat easier on the cables.

The digging ladder is usually of plate girder design and this shape is by far the best. It is subjected to cross bending stresses and in addition must act as a column. Also, when the dredge is swinging there is a tendency to buckling due to the tension caused by the bow lines. The depth of the girder at the center is usually from 1/10 to 1/12 of the length. The stresses can be computed from the strain sheet. Usually the fiber stress in the tension side is limited to 13,500 lb. per sq. in. and 15,000 lb. per sq. in. on the compression side. The material is ordinary structural steel having the chemical and physical characteristics of the material used in railway bridge construction.

The riveting must be very well done and the splices are usually

designed to have 25 per cent excess strength over the solid sections.

The stacker is what is ordinarily known as a Warren girder. It is pivoted on the stern of the dredge and supported by tackle at two points. In determining the stresses it is drawn in skeleton outline at an angle of 18 deg. with the surface of the deck, as this angle is the greatest at which boulders can be elevated by conveyor belts without excessive slipping and rolling. The same grade of structural material is used in its construction as in the digging ladder and the same care must be used in supplying it with ample gusset plates at the panel points and with good riveting.

Dredges run 24 hours a day every day in the year except Christmas and the Fourth of July, or when operations cease due to cleaning up or on account of repairs. Well designed dredges have a running efficiency as high as 80 per cent over a period of 12 months. This means that gravel is actually passing through the machine and being washed during 80 per cent of each 12 months' period. When the hard duty, excessive wear and racking strains of these machines are considered, it speaks well for their designers and operators.

S. L. G. Knox (written). It is the writer's opinion that the materials commonly used in bucket bottoms are cheap per yard handled in direct proportion to their cost per pound; that is, carbon steel, cheapest per pound, is most expensive in wear and breakage; nickel and nickel chrome come next, and probably manganese steel is the cheapest per yard, although the latter has not yet been used widely enough for bucket bottoms to justify final conclusions. Teeth on the lips could be used to excellent advantage in some cases—not only where the digging of excessively hard gravel or bedrock would be facilitated thereby, but also where sticky clay is encountered. In the latter case there is no difficulty in digging to be overcome by the use of teeth, but the teeth break up the material entering the bucket in such a way as to prevent it from lying close and solid against the hood; this squeezes out the air, and so prevents the material from dumping.

Among the frequent and troublesome renewals on a placer dredge are the upper tumbler plates. These must be firmly bolted, as they are subjected to constant and heavy blows from the bucket

chain, and as the nuts are unavoidably difficult to get at much delay results. On some dredges now being built the bolts have key slots and double keys instead of nuts, so that a blow of a hammer will either tighten or loosen the bolt as may be desired.

Although there is an advantage in using a 7 or 8-sided lower tumbler, the writer sees no virtue in a 7 or 8-sided upper tumbler, as it would be difficult to secure a good driving grip on the bucket chain, and as the weight not only of the tumbler, but of the tumbler shaft, gears and bearings of the whole main drive would be proportionately increased. In the largest and best elevator dredges for other than placer dredging, there are rarely if ever more than five sides to the upper tumbler, and, considering the great saving in cost of the whole driving system, which is one of the heavy units of the dredge machinery, this practice has much to commend it.

The writer does not agree with the author's comments on the disadvantages of concentric support of the digging ladder (Par. 53). Several different methods of accomplishing this have been worked out, and it is possible to secure the operating advantages of this arrangement without making it more difficult to remove the upper tumbler and shaft, and with a decrease of weight and cost, as compared with the independent suspension. A new method of accomplishing this which requires no bearing castings, and does away with the eccentric stresses on ladder ends and gantry brackets, calling for excessive weight and cost in some of the designs previously adopted, is being installed on several large dredges now under construction.

To overcome such difficulties as unavoidable inaccuracies of pitch in the large gearing required for driving the upper tumbler shaft, distortion of the pulley shaft due to the necessity of driving the gearing by a belt or pulley which lies outside the gear train, and therefore transmitting all the load through a short length of shaft to the near pinion, and half the load to the remote pinion, there has been introduced into the design of some recent large dredges differential pulley shafts, working on the principle of the differential in the rear axle of an automobile, but without gears, as a simple system of levers and sliding blocks gives all the motion required. In this way the power is always divided exactly between two trains of gearing, regardless of any inaccuracies of pitch or torsional yield of shafting, and no portion of the pulley

shaft is supported by more than two bearings, which is the ideal arrangement when a perfectly stable foundation is impracticable.

In some recent dredges the usual type of save-all grizzly has been radically departed from. The material, instead of falling against a screen of bars arranged closely together, and set at a steep angle, so that much of the material slides off into the well hole without reaching the sluice, falls upon a stout table having only enough slope from the horizontal to permit the material to slide forward over a set of grizzly bars which reject the large stuff into the well hole and drop the fine material onto a series of save-all sluices.

It seems to be the common belief that two equal spuds should be provided, so that if one breaks, there will be another available to carry on the work while the injured one is being repaired. The writer sees no reason why spuds should break more than digging ladders or bow gantries, and states that whenever they have been properly designed, they have stood up to their duty as surely as any other part of the dredge. In figuring the possible stresses in some of the spuds that have broken the writer has proved them to go up to 40,000 lb. and 60,000 lb. per sq. in., which explains why they broke.

One cause of bending spuds often overlooked results from swinging the dredge after one corner has come in contact with the tailings pile. This may result in bending the spud backward, but may be overcome by limiting the strength of the lower spud keepers, so that they will pull out under a less outward force than will bend the spud.

In concluding the writer called attention to the conservatism of dredging companies, who have now at their command a very satisfactory instrument, and are rather slow in adopting more or less radical changes.

Mr. Knox told further of a novel method of checking the stress diagram. He laid a Bristol recording gage on its side, connected a reach rod with the spud, let the pencil rest on the revolving dial, and kept it running for 48 hours. As the dredge yielded back and forward under the pressure at the point of the digging ladder, it moved the pencil out and in, and an exact record was made on the revolving disk. The springs were tested by the maker when they were turned out, and there was an exact record of the number of pounds that would compress each spring

an inch, so that the difference between the point of support and the leverages was known, and in that way an automatic record was obtained, the first ever made of the actual stresses in the dredge. It was this which led the writer to put the dredge spud in the center on the dredges they are now building, using a single spud. That divides the stress equally between two main trusses, takes all torsion stresses out, and saves weight in the hull. There is of course a walking spud which has about five-eighths of the stress of the other spud.

Charles Whiting Baker explained why the ladder dredge is practically the only one of the four existing types of dredges suited to gold dredging. Both the clamshell and the dipper dredges let out part of the water and mud as the bucket comes up, and with it some of the gold. As to the hydraulic dredge, it is best adapted to clay and sand, while gold is usually found in rather coarse material. It is true that hydraulic dredges have been developed so that they will work satisfactorily with occasional boulders the size of a man's head and sometimes larger, but on the whole there is every reason to believe that the ladder dredge is the type destined to be used prmanently for the saving of the gold in gravel.

A dredge of that sort, in order to be efficient, has to be a pretty fair-sized machine, since otherwise the capacity of any one single bucket is too small to handle a large size cobblestone; on the other hand a large size bucket means a large and expensive machine.

There are many places where it would be desirable to operate a dredging plant on a small scale, and the present large, heavy and expensive ladder dredge is not suitable for such work; there may be, however, special kinds of dredging machinery for use in the case of smaller undertakings, which could have a bucket closed at the back, and not open like an ordinary dipper dredge, and could handle good paying ground to much advantage.

To give an idea of the different types of dredges used at the present time, Mr. Baker described the work on the Panama Canal, and the latest dredge down there. The old French excavating machine used in the Culebra cut was an endless chain bucket excavator which is one of the finest machines to do work with where there is granular earth which handles well, such as sand, or

gravel, but which is extremely inconvenient for digging wet clay which sticks in the buckets and stays there. On the Panama Canal, however, clay and rock are the two classes of materials that one has to deal with. It is quite clear, with the experience that has now been gained, that the Culebra Cut could never have been successfully dug with that type of machine, because it is not suited to handle either clay or broken, blasted rock. When the steam shovel was invented, it was not expected to handle anything but ordinary dirt, but it can now handle anything that can be balanced across the bucket.

Mr. Baker further described other kinds of dredges used on the Panama Canal, and expressed the general view that while excellent work had been done by American engineers in developing the dipper type of excavator, there was a little too much inclination in this country to neglect the possibilities of the ladder excavator.

E. Gybbon Spilsbury described another type of gold dredge used in the placers of Siberia which is somewhat on the type of the dipper dredge, but with a closed bucket. It is a comparatively cheap machine to build. It works from the end of a crane boom having a large radius, and is dropped down in one direction, and a lead rope drags it through the mud or gravel; then it reverses and brings up and discharges the material on gold-saving tables. These dredges have done remarkably good work and been very profitable to operate. Answering an inquiry, Mr. Spilsbury stated that there is no method of cleaning the surface of the rock more thoroughly, either by suction, water jet, or otherwise, supplementary to the work of the ladder dredge, but the modern dredges are so powerful that, as a matter of fact. they cut from three to six inches off the bed rock in nearly all cases. The bed rock is more or less soft, especially in California, and the dredge actually cuts enough from its surface to be sure that they get everything lying on the bed rock.

John M. Sherrerd added that the reason why these dredges are able to go through the six inches or one foot of rock or slate is because of the use of manganese steel lips. The latest idea in this direction is to make the whole bucket of manganese steel. Such

buckets seem to be very successful, and possibly they will be regularly used for this purpose in the future. Those made up to the present time run 5, 9, and even 12 and 13 cu. ft. capacity.

H. S. Wood (written) could not endorse a six-sided top tumbler. Practically all English ladder dredges have four-sided top tumblers. This made the chain slap so much by the lift of the tumbler corners that Mr. A. W. Robinson sought to improve the possible speed of the chain by making the top tumbler five-sided. The improvement was marked, but at least once the top tumbler slipped around one link space (72 deg.) under the chain without carrying the chain with it, the digging bucket having caught under a boulder. If a five-sided tumbler permits this, a six-sided one will be much more unreliable in hard digging.

The author presented no closure.—Editor.

No. 1349

PROBLEMS IN NATURAL GAS ENGINEERING

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Member of the Society

The extensive increase in the production and distribution of natural gas in recent years has given rise to engineering problems of more or less unique interest. For a considerable period after natural gas had become an important factor in industrial life it was handled almost entirely by rule of thumb methods, but with the advent of large compressing stations and the enforced realization that the gas supply is not by any means unlimited, the application of engineering principles became essential, not only to secure greater economy, but also to cope successfully with the growing complexity of the problems encountered. The object of this paper is to present the most important of these problems and to outline the methods used in solving them. First, however, a brief account of the characteristics of natural gas as it is found in America, may not be out of place.

PROPERTIES OF NATURAL GAS

2 *Origin and Composition.* Natural gas and petroleum are closely related and it is generally believed that they have been generated in the earth by the same process. What this process is, or may have been, is a question of pure speculation among scientists, two principal theories having been advanced to explain it, both based upon geological observations.

3 The first hypothesis, the so-called chemical, or inorganic theory, holds that the gas is constantly being formed by the action of water on the carbides of different metals in the far inner regions of the earth, thus producing the hydrocarbons forming the principal constituents of natural gas. These hydrocarbons are then forced outward toward the earth's surface by the high pressures generated until they finally lodge in the porous rocks where they are found.

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4 The second, or organic theory, maintains that the hydrocarbons were formed by the partial decomposition under water, of animal or vegetable matter in the rock strata where the gas is now found.

5 There are many arguments in support of each theory, as it is known that hydrocarbons can be produced by either of the methods suggested. This being so, it seems reasonable to suppose that both agencies were at work and that each may claim its share of the credit for the production of the vast quantities of gas obtained throughout the world.

6 By whatever manner produced, however, the fact remains that hydrocarbons form the principal constituents of natural gas as found in America, although other gases are present in widely varying, but relatively small proportions.

7 The analyses of gases found in pools covering practically the whole productive area of the United States are given in Table 1, which was compiled from various sources, principally the reports of the Geological Survey of West Virginia,[1] and of Kansas.[2] These reports give very complete discussions of the theories mentioned, and describe in detail what the author believes to be the most reliable analytical work done on natural gas, namely, that of Prof. F. C. Phillips on Pennsylvania Natural Gas, in 1887, and that of Cady and McFarland on the Gases of Kansas, in 1906.

8 The table shows the apparent lack of uniformity existing in the constitution of natural gas from different fields; but the most noticeable feature is the great preponderance of hydrocarbons, especially of the methane, or paraffin series. This group has the general chemical formula C_nH_{2n+2}, its chief member being methane, or marsh gas (CH_4). Methane occurs in all natural gas in proportions varying from about 14 per cent up to 97 per cent. It possesses high heating value but has practically no illuminating properties.

9 The second member of the series, ethane (C_2H_6), occurs in percentages ranging from 0 up to 40 or more. This gas is of much higher fuel value than methane, is considerably heavier, and possesses greater illuminating properties. Thus the heating value, specific gravity, and illuminating value of the gas in which it occurs are all raised by the presence of ethane.

10 The higher members of this series, propane (C_3H_8) and butane (C_4H_{10}), are present in dry gas only in minute quantities, if at all, and they are consequently of small importance.

[1] Vol. 1-a, 1904.
[2] Vol. 9.

11 Of the olefine series, or so-called "illuminants," having the general formula C_nH_{2n}, the only member present in measurable quantity is the leading one of the group, ethylene (C_2H_4), and this only in a fraction of 1 per cent.

12 Carbon monoxide (CO) is occasionally reported, although in such small proportion as to add very little to the fuel value of the gas.

13 Hydrogen has been reported in many analyses of natural gas, but, as has been pointed out by Phillips, Cady and McFarland, these indications were probably due to erroneous deductions from experimental results, and it is extremely doubtful if free hydrogen is ever present.

14 Hydrogen sulphide (H_2S) is found in the natural gas from a few isolated areas, but is not at all general. Its presence is indicated by a distinctive odor. At most, it exists in but a small fraction of 1 per cent and is, therefore, relatively unimportant.

15 The above list comprises all the gases ever found in natural gas that contribute to its heating value, and it will be seen that the only ones of any practical importance are methane and ethane, the first two members of the paraffin series.

16 In addition to the above, there are always found a number of inert gases, chief of which is nitrogen (N), ranging from less than 1 per cent to almost 83 per cent in the various gases as shown in Table 1. The majority of natural gases, however, that exist in any considerable quantity, or come from any great depth, seldom exceed 10 per cent in nitrogen content, and usually range from 4 to 7 per cent. Nitrogen is a non-reactive, elementary gas that acts merely as a diluent of the natural gas, reducing its fuel value.

17 Next in importance is carbon dioxide (CO_2), which is found in proportions ranging from 0 up to about 4 per cent. It acts in the same manner as nitrogen, by reducing the effective heating value of the gas.

18 Oxygen has frequently been reported in varying quantities, but the probability is that it is present in traces only, and that the apparent high percentages sometimes reported were due to the samples having become contaminated by air.

19 The work of Cady and McFarland has shown that in all probability the element helium is a constituent of nearly all natural gases, together with other members of the argon group. Helium was

TABLE 1 ANALYSES OF GASES FROM THE WHOLE PRODUCTIVE AREA OF THE UNITED STATES

Location (Kansas Gases)	Year	O_2	CO_2	C_2H_4	CO	CH_4	C_2H_6	$\frac{CH_4}{C_2H_6}$	H	H_2S	He	N_2	B.T.U. by Analysis	B.T.U. by Equation [4]	Specific Gravity
*Dexter	1905	0.20	0.00	.....	0.00	14.85	0.41	15.22	trace	.....	1.84	82.70	139.7	82.6	0.8935
*Dexter (Greenwell Well)	1906	0.10	0.00	0.00	0.00	14.33	1.06	15.39	trace	.....	1.64	82.87	245.5	89.6	0.8977
*Eureka, new field	1906	0.10	0.20	0.00	0.00	51.80	0.00	51.80	0.00	.....	1.50	46.40	465.0	427.6	0.7430
*Eureka, town supply	1906	0.50	0.20	0.00	0.00	51.40	0.00	51.40	0.00	.....	1.50	46.40	461.0	424.6	0.7451
*Fredonia	....	trace	0.61	0.12	0.00	82.25	0.00	82.25	0.00	.....	0.616	16.40	739.8	729.6	0.6253
*Elmdale	1906	0.30	0.15	0.55	0.00	78.60	7.71	86.31	0.00	.....	0.56	12.13	836.1	826.1	0.6441
*Moline	1906	0.00	0.54	.....	0.00	74.10	0.00	74.10	trace	.....	0.51	24.85	665.0	652.2	0.6599
Burlington	1906	0.00	0.00	0.20	0.00	85.50	3.20	88.70	0.00	.....	0.495	10.60	820.9	810.3	0.6113
New Albany	1906	0.00	0.20	0.25	0.00	89.10	0.00	89.10	0.12	.....	0.49	9.84	804.0	794.7	0.5944
*Lawrence, deep well	1906	trace	0.92	0.00	0.00	81.40	0.00	81.40	0.00	.....	0.46	17.22	730 0	723.9	0.6318
*Olathe	1906	trace	0.00	0.10	0.00	84.40	0.00	84.40	0.00	.....	0.40	15.10	758.5	746.2	0.6148
Garnett	1906	trace	0.20	0.16	0.00	94.30	0.36	94.66	0.00	.....	0.37	4.61	855.1	848.9	0.5748
*Eudora	....	0.31	0.62	0.00	0.00	88.60	0.00	88.60	0.00	.....	0.27	10.20	795.0	792.3	0.6021
Parsons (Wilson County Gas)	....	trace	0.72	0.00	0.00	91.90	3.37	95.27	0.00	.....	0.27	3.74	878.8	877.7	0.5902
Buffalo	1906	trace	0.00	0.00	0.11	96.20	0.78	96.98	0.18	.....	0.27	2.46	876.3	873.5	0.5656
Altoona	1906	0.00	0.92	0.00	0.00	92.00	2.85	94.85	0.00	.....	0.263	3.97	870.5	873.1	0.5911
*Augusta	1906	0.00	0.00	0.77	0.00	79.10	7.44	86.54	0.00	.....	0.25	12.44	840.0	831.4	0.6430
Chanute	1906	0.10	0.00	0.00	0.00	94.70	0.00	94.70	0.00	.....	0.24	4.96	850.0	844.7	0.5731
Moran	1906	0.20	0.30	0.35	0.20	92.00	0.00	92.00	0.39	.....	0.214	6.35	832.9	831.8	0.5829
Peru	1906	0.10	0.51	0.00	0.00	81.70	7.60	89.30	0.00	.....	0.19	9.39	854.2	852.5	0.6309
Iola	1906	0.23	0.00	0.00	0.00	94.50	0.00	94.50	trace	.....	0.183	5.08	848.0	843.9	0.5746
Lawrence pipe line, April 16	1906	trace	0.20	.....	0.00	98.06	0.00	98.06	trace	.....	0.17	1.57	880.0	878.3	0.5605
Pipe line, October 23	1905	0.12	0.00	.....	0.00	98.00	0.00	98.00	0.00	.....	.....	1.88	879.0	879.0	0.5615
Pipe line, December 12	1906	0.24	1.94	0.00	0.00	94.30	0.75	95.05	0.00	.....	0.17	2.60	858.0	869.8	0.5868
Arkansas City	....	0.20	0.10	0.10	0.00	81.10	11.95	93.05	0.00	.....	0.159	6.39	920.0	915.8	0.6392
Blackwell, Okla	1906	0.00	0.00	0.61	0.00	83.40	10.31	93.71	0.33	.....	0.16	5.19	922.4	918.4	0.6247
Humboldt	1906	0.10	0.00	0.81	0.00	94.00	1.97	95.97	0.00	.....	0.14	5.19	887.3	880.8	0.5996
Iola	1906	0.40	0.70	0.00	0.30	91.50	0.00	91.50	0.00	.....	0.132	6.97	822.0	826.8	0.5917
Erie	1906	0.22	0.33	0.30	0.00	90.30	4.26	94.56	0.00	.....	0.13	4.45	882.3	881.8	0.5971
Bonner Springs	....	0.10	0.00	0.00	0.00	97.18	0.00	97.18	0.25	.....	0.104	2.36	871.7	870.4	0.5616
Altamont pipe line, Liberty and Coffeyville	....	0.00	0.92	0.61	0.00	95.70	0.00	95.70	0.00	.....	0.08	2.69	868 0	872.1	0.5751
Caney	....	0.15	0.81	0.10	0.00	92.40	0.00	92.40	0.00	.....	0.08	6.46	830.5	833.1	0.5886
Sheffield, Mo	....	0.20	0.83	0.50	0.10	92.90	0.00	92.90	0.00	.....	0.041	5.43	840.7	845.9	0.5869
Kansas City, Mo., 2416 Tracey Avenue	....	0.10	0.60	1.20	0.20	87.20	7.03	94.23	0.00	.....	0.013	3.65	912.2	917.8	0.6141
Paola	1906	0.40	0.70	0.00	0.00	98.00	0.00	98.00	0.00	.....	0.0093	0.88	879.0	884.7	0.5655
Topeka pipe line, December 22	1907	0.20	1.53	0.00	0.00	94.56	0.00	94.56	0.00	.....	.....	3.71	849.0	858.1	0.5841
Lawrence pipe line, December 23	1907	trace	1.85	0.00	0.00	94.47	0.00	94.47	0.00	.....	.....	3.68	848.0	859.6	0.5861
Gas from coal mine, Scranton, Pa	....	0.92	1.06	0.39	0.13	94.2	0.00	94.20	0.00	.....	trace	3.31	851.1	861.0	0.5842
Lawrence pipe line, July 11	1908	0.25	0 81	0.38	0.81	93.50	0.00	93.50	0.00	.....	.....	4.25	847.2	877.6	0.5848
*W. Bloomfield, N. Y	1870	0.23	10.11	2.94	....	82.41	.....	.....	....	.....	4.31	.....	782.3	855.6	0.6819

*Olean, N. Y.	1885	2.00	.....	1.00	0.50	96.50	.....	.....	....	.....	.....	.....	881.3	890.8	0.5703
Fredonia, N. Y.	1887	....	0.41	.....	....	46.43	43.62	90.05	....	.....	.....	9.54	1102.5	1110.2	0.8078
Baldwinsville, N. Y., (Trenton)	1899	trace	.....	0.25	0.95	98.40	.....	98.40	trace	.....	.....	0.40	889.3	900.4	0.5596
Morgantown, W. Va. (Big Injun)	1904	0.20	0.006	0.40	0.40	80.94	14.60	95.54	trace	.....	.....	3.46	965.4	969.1	0.6427
Fairmount, W. Va. (Bayard)	1904	0.20	0.100	0.20	0.40	81.60	14.09	95.69	0.20	.....	.....	3.21	961.2	965.6	0.6379
Shinnston, W. Va. (Big Injun)	1904	0.20	0.000	0.40	0.40	79.95	15.09	95.04	0.00	.....	.....	3.96	964 7	968.5	0.6469
Shinnston, W. Va. (Gordon)	1904	0.10	0.00	0.20	0.40	80.85	14.88	95.73	0.10	.....	.....	3.47	966.7	969.6	0.6420
Lumberport, W. Va. (5th. sand)	1904	0.30	0.10	0.10	0.40	80.70	14.35	95.05	0.10	.....	.....	3.95	955.8	958.6	0.6430
Shinnston, W. Va. (50 ft.)	1904	0.30	0.00	0.20	0.50	86.48	7.65	94.13	0.00	.....	.....	4.87	902.0	905.9	0.6148
Sheffield, Pa	1887	trace	0.30	.....	....	70.55	20.09	90.64	....	.....	.....	9.06	953.1	951.4	0.6908
Kane, Pa	1887	trace	0 20	.....	....	68.96	21.05	90.01	....	.....	.....	9.79	954.0	950.6	0.6975
Wilcox, Pa	1887	trace	0.21	.....	....	72.64	17.74	90.38	....	.....	.....	9.41	934.5	931.2	0.6803
Speechley, Pa	1887	trace	0.05	.....	....	67.89	27.55	95.44	....	.....	.....	4.51	1048.0	1046.3	0.7057
Lyons Run, Murraysville, Pa	1887	trace	0.28	.....	....	97.60	0.10	97.70	....	.....	.....	2.02	877.1	877.2	0.5647
Raccoon Creek, Pa	1887	trace	trace	.....	....	73.72	16.37	90.09	....	.....	.....	9.91	922.0	918.9	0.6745
Baden, Pa	1887	trace	0.41	....	....	70.66	16.61	87 27	....	.....	.....	12.32	885.5	882.4	0.6879
Houston, Pa	1887	trace	0.44	.....	....	65.71	18.55	84 26	....	.....	.....	15.30	899.0	895.5	0.7109
Murraysville, Pa	1894	trace	0.20	.....	....	93.33	2.07	95.40	....	.....	.....	4.40	870.0	868.7	0.5833
Pittsburg, Pa., (Exhibition Grounds)	1894	trace	0.52	.....	....	87.56	4.62	92.18	....	.....	.....	7.30	859.1	860.1	0.6108
Creighton, Pa	1894	....	3.64	.....	....	86.87	9.49	96.36	....	.....	.....	0.00	930.8	957.5	0.6340
Pittsburg, Pa. (Painter Well)	1894	....	0.40	.....	....	92.67	6.23	98.90	....	.....	.....	0.70	930.9	933.7	0 5900
Allegheny City, Pa. (Salt Well)	1894	....	0.30	.....	....	87.53	5.07	92.60	....	.....	.....	7.10	866.4	865.5	0.6101
Gambier, O	1875	0.80	0.30	.....	0.50	81.40	12.20	93.60	....	0.650	.....	4.80	924.5	932.7	0.6414
Findlay, O	....	0.34	0.26	0.30	0.50	92.61	.....	92.61	2.18	0.20	.....	3.61	844.1	851.7	0.5665
Thurston, O. (Clinton)	....	0.15	0.25	0.30	0.15	90.48	.....	90.48	0.55	.....	.....	8.12	817.9	818.3	0.5894
Fostoria, O	....	0.35	0.20	0.20	0.55	92.84	.....	92.84	1.89	0.15	.....	3.82	843.7	851.1	0.5678
Findlay, O	....	0.39	0.25	0.35	0.41	93.35	.....	93.35	1.64	0.20	.....	3.41	849.6	856.2	0.5682
St. Marys, O	....	0.35	0.23	0.20	0.44	93.85	.....	93.85	1.74	0.21	.....	2.98	852.0	859.7	0.5652
Cleveland, O	1894	....	0.20	.....	....	76.77	16.73	93.50	....	.....	.....	6.30	955.6	953.5	0.6620
Butler, O	....	0.05	0.00	0.40	0.00	70.00	16.75	86.75	0.27	.....	0.15	12.38	901.6	895.0	0.6857
Muncie, Ind. (Trenton)	....	0.35	0.25	0.25	0.45	92.67	.....	92.67	2.35	0.15	.....	3.53	844.5	850.9	0.5647
Anderson. Ind. (Trenton)	....	0.42	0.26	0.49	0.73	93.07	.....	93.07	1.86	0.15	.....	3.02	850.5	860.7	0.5676
Kokomo, Ind. (Trenton)	....	0.30	0.29	0.30	0.55	94.16	.....	94.16	1.42	0.18	.....	2.80	856.2	864.4	0.5668
Marion, Ind. (Trenton)	....	0.55	0.30	0.15	0.60	93.58	.....	93.58	1.20	0.20	.....	3.42	848.6	858.8	0.5720
Marion, Ind. (Trenton)	....	0.00	0.73	0.86	0.00	77.40	14.18	91.58	0.00	.....	0.167	6.66	933.2	933.1	0.6592
Kokomo, Ind. (Trenton)	....	.. .	0.40	.....	....	88.91	4.69	93.60	....	.....	.....	6.00	871.3	872.0	0.6046
Oregon	1904	....	2.40	.....	....	88.10	.....	88.10	....	.....	.....	9.50	790.5	804.3	0.6159
*Fort Worth, Tex	1910	0.20	0.00	0.30	0.30	55.90	5.50	61.40	1.00	.....	.....	36.80	596.8	585.6	0.7320
Point Albino, Canada	1896	....	trace	.....	....	85.83	10.74	96.57	....	0.74	.....	2.69	945.8	953.2	0.6208
Vancouver, B. C	....	....	0.14	.....	....	.....	.....	93.56	....	.....	.....	6.30	.....	.....	.....
Bottineau, N. D	....	3.00	.....	0.20	1.20	82.70	.....	82.70	0.50	.....	.....	12.40	750 2	763.4	0.6246
Kenova, W. Va	....	0.60	0.30	0.70	0.30	78.30	19.60	97.90	....	.....	.....	0.20	1026.2	1036.8	0.6621
Kenova, W. Va	....	0.40	0.20	0.70	0.60	78.90	18.80	97.70	....	.....	.....	0.40	1019.5	1029.2	0.6551

found in the Kansas gases in percentages as high as 1.8, and in only one case of all the gases tested was none found. Inasmuch as no effort was made to detect this gas in any former analyses, it is probable that it was present, although undetected, since it was usual to consider as nitrogen all residue left after determining the other gases present.

20 From the above it will be seen that while natural gas may contain a considerable number of constituent gases, the great bulk of it is composed principally of but four or five of any importance.

21 In Table 1 the gases marked with an asterisk are samples that came from small shallow wells, or were otherwise abnormal, and thus cannot be considered as representative. Excluding these from the list and striking an average of those remaining, the constitution of what may be termed the average natural gas, after slight modifications in accordance with the above discussion, would be about as in Table 2.

TABLE 2 CONSTITUTION OF THE AVERAGE NATURAL GAS

Methane (CH_4)	87.00
Ethane (C_2H_6)	6.50
Ethylene (C_2H_4)	0.20
Carbon monoxide (CO)	0.20
Hydrogen (H)	trace
Nitrogen (N_2)	5.50
Carbon dioxide (CO_2)	0.50
Helium (He)	0.10
Oxygen (O_2)	trace

B.t.u. per cu. ft. at 29.82 in. and 60 deg. = 887.3.
Specific gravity = 0.6135.

22 *Heating Value of Natural Gas.* The fuel value of natural gas depends principally upon the relative proportions present of methane, ethane, and the inert gases, such as nitrogen and carbon dioxide. The only satisfactory manner of ascertaining the heating value is by means of calorimetric determinations, but as very few of these were reported with the analyses shown in Table 1, and in order to arrive at some comparative idea of the relative fuel values of the various gases listed, column 14 is given, showing the b.t.u. per cu. ft. as computed from the analyses. In making calculations, the constants in Table 3 were used.

23 In the discussion of properties of gases it is necessary to refer all volumes to some definite standards of pressure and tem-

perature. The standards most used in scientific work are 29.92 in. (760 mm.) of mercury, corresponding to 14.7 lb. per sq. in. for the pressure base, and 32 deg. fahr. (0 deg. cent.) for that of the temperature, while in industrial work 30 in. of mercury and 60 deg. fahr. are largely used. Some years ago, F. H. Oliphant, geologist for the Standard Oil Company, published a handbook on natural gas, in which he established the pressure base of 14.65 lb. per sq. in. absolute, and temperature base of 60 deg. fahr., and since that time it has been customary for natural gas men to refer their measurements to these bases. A pressure of 14.65 lb. per sq. in. is 4 oz. above an average assumed atmospheric pressure of 14.4 lb., the latter being the average at about the elevation of the Great Lakes, which elevation was considered as fairly representing that of most of

TABLE 3 HEATING VALUE AND SPECIFIC GRAVITY OF THE GASES FOUND IN NATURAL GAS

Kind of Gas	Symbol	Net Heating Value, B.T.U. per Cu. Ft.	Specific Gravity Air = 1.0
Methane	CH_4	897	0.5529
Ethane	C_2H_6	1594	1.0368
Ethylene	C_2H_4	1471	0.9676
Carbon monoxide	CO	322	0.9671
Hydrogen	H	271	0.0692
Hydrogen sulphide	H_2S	615	1.1769
Nitrogen	N_2	....	0.9701
Carbon dioxide	CO_2	....	1.5195
Helium	He	....	0.1382
Oxygen	O_2	....	1.1052

the gas fields. Thus, 4-oz. gas is the generally accepted pressure unit of measurement among natural gas men and will be so considered throughout this paper. Frequently gas is bought or sold on different pressure bases, and in such cases the corresponding absolute pressures are determined by considering atmospheric pressure as being equal to 14.4 lb. per sq. in., as for instance, a 10-oz. selling base corresponds to 14.4+0.625=15.025 lb. absolute.

24 Conforming with this usage, therefore, all formulae, heating values, etc., throughout this paper are given for the unit of measurement as being 1 cu. ft. at 60 deg. fahr. and 14.65 lb. per sq. in. absolute (29.82 in. of mercury), which is commonly referred to as "gas standard."

25 It will be noted in Table 1 that the heating values given are

the net or low values. In publishing gas analyses or constants, many writers, in fact, most of them, give the high values, but the author believes the low to be more rational, inasmuch as in nearly all uses to which natural gas is put as a fuel, it is impossible to recover the latent heat of vaporization of the steam formed by the combustion, and therefore the net heating values give a more truthful conception of the actual worth of the gas for fuel purposes.

26 When it is impossible to obtain a calorimetric determination of the heating value of a particular gas, the next best procedure is to compute it from the chemical analysis of the gas, using the values shown in Table 3 for the heating values of the constituent gases. This, of course, is done by multiplying the percentage of each gas present by its corresponding heating value per cubic foot, and adding the products. The specific gravity is obtained by computation in precisely the same manner. Such computed results are necessarily subject to whatever errors there may be in the analysis of the gas, and unless this has been done with great care and precision, a wide discrepancy may exist between the calculated and actual values.

27 It is oftentimes desirable to gain an approximate knowledge of the heating value of a gas when neither a calorimetric determination nor a chemical analysis is available. In such cases, a fair "guess" may be made from a determination of the specific gravity of the gas, provided it is known to be a normal "dry" gas without freakish tendencies. The specific gravity is readily determined by the effusion method, in which the time required to pass a given quantity of the gas through a pin-hole orifice in a thin plate under a given head, or pressure, is divided by the time required to pass the same quantity of air through the same orifice and under the same pressure; the square of the quotient being the specific gravity of the gas, referred to air. For most reliable results the air and the gas should be run at the same temperature, to avoid the necessity of a correction for this factor.

28 An approximate formula for determining the heating value from the specific gravity may be derived from the following considerations. In the analysis given in Par. 21 for an average natural gas, and which represents the average constitution of the gases considered as representative in Table 1, it will be seen that of the combustible gases, methane and ethane, comprise 93.5 per cent of the whole and ethylene and carbon monoxide comprise 0.2 per cent each. No great error will be made, therefore, if these two latter gases are con-

sidered as being a part of the paraffin group, especially since ethylene and ethane do not differ greatly in either heating value or specific gravity. The inert gases may likewise be combined in one group, of which the resultiug specific gravity may be considered equal to 1.0. Consequently, for approximate results, the average natural gas may be regarded as made up of three distinct gases, methane, ethane, and "inerts," of which the heating values and specific gravities may be considered as in Table 4.

29 Representing the relative volumetric proportions of these gases, expressed decimally, as m, e, and i, respectively, the following relations will obtain:

$$m+e+i=1.0 \quad [1]$$
$$H=897m+1594e \quad [2]$$
$$G=0.5529m+1.0368e+1.0i \quad [3]$$

in which H = the lower heating value in b.t.u. per cu. ft. of natural gas at gas standard, and G = the specific gravity of the gas. Eliminating m and e from equations [1], [2], and [3], the heating value of the gas may be expressed in terms of i and G as

$$H=1440G-1541i+100.6 \quad [4]$$

30 The sum total of the percentages of the inerts given in Par. 21 is $0.061=i$. Substituting this in equation [4]

$$H=1440G+6.6 \quad [5]$$

Applying equation [5] to the average gas of Table 2, of which the

TABLE 4 HEATING VALUE AND SPECIFIC GRAVITY OF METHANE, ETHANE AND INERTS

Kind of Gas	B.t.u. per Cu. Ft.	Specific Gravity
Methane	897	0.5529
Ethane	1594	1.0368
Inerts	....	1.0000

specific gravity is 0.6135, it would indicate the gas to have a fuel value of $H=890$ b.t.u. per cu. ft., as compared with a value of 887.3 as computed from analysis.

31 The values of H and G given in Table 1, when plotted against each other, yield the result shown in Fig. 1. In this figure, only those gases considered as representative, and not marked with an asterisk, have been plotted. The resulting points are very widely scattered, but the general tendency is toward an increasing value of H with increasing G, as would naturally be expected, and it must be remembered that many of the analyses plotted cannot be vouched

for as to accuracy; in fact it is reasonably certain that they are inaccurate. The diagonal line drawn in the figure is the graph of equation [5], and is not far from an average line through the points shown. It is fair to conclude, therefore, that for purely approximate work, a reasonable notion of the heating value of natural gas may be obtained from the known specific gravity by using equation [5]. When the locality is known from which the gas is derived, the approximate proportion of the inert gases may be learned by selecting, from Table 1, a gas from the same field, and the heating value com-

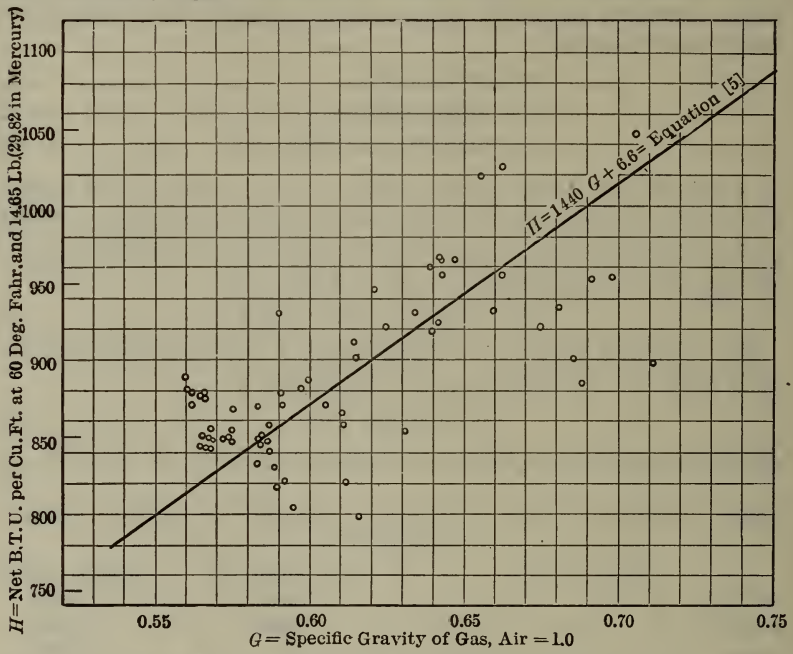

FIG. 1 CURVE SHOWING RELATIONS OF HEAT CONTENT AND SPECIFIC GRAVITY OF GAS, FROM TABLE 1

puted by means of equation [4]. The values calculated in this way are shown in column 15 of Table 1, and may be compared with those computed directly from the analyses.

32 As stated, no great degree of accuracy is claimed for this method of computation, but the author has frequently found it of great convenience where other and better methods were not available. The comparative values shown in the table as computed from the analyses and by equation [4], indicate that a fair degree of approximation can be attained by it.

33 Reference has been made to "dry" gas. This term is used to designate the ordinary natural gas coming from a well producing gas exclusively, as distinguished from that coming from an oil well, which is called "wet." These terms have no reference to water vapor in the gas, but imply the absence or presence of vapors of hydrocarbons higher in their respective series than those discussed above.

TRANSMISSION OF NATURAL GAS

34 *Pipe-Line Flow Formula.* In the design of pipe lines for the transmission of natural gas from the field to the points of consumption it is necessary to make use of a formula expressing the relations to each other of the quantity and initial and final pressures of the gas, and the diameter and length of line. Many such formulae have been proposed giving widely differing results. In nearly all of them the flow is stated as varying as the square root of the fifth power of the pipe diameter, and either the coefficient of friction is considered constant, or a different coefficient is given for each diameter of pipe. This serves well enough where the diameter is known and any one of the other quantities expressed by the formula is desired, but is somewhat awkward when it is desired to ascertain the diameter of line necessary to meet the other given conditions.

35 The author has derived a new formula, which he believes expresses the relationship of the quantities involved even more closely than any heretofore offered. It is based on isothermal flow, and the variation in the value of the coefficient of friction is provided for without complicating the formula, yet permitting the required diameter of line to be ascertained readily.

36 The expression for the initial velocity of any gas flowing in a pipe is given by Unwin [1] as

$$u_1 = \sqrt{\frac{gCTm}{fl}\,\frac{(P_1^2 - P_2^2)}{P_1^2}} \qquad [6]$$

where

u_1 = initial velocity, ft. per sec.

g = acceleration due to gravity

C = thermodynamic constant of the flowing gas $= \frac{PV}{T}$

T = absolute temperature of gas

m = hydraulic mean radius of the pipe $= \frac{D}{4}$

[1] Compressed Air, p. 67, Van Nostrand's Science Series.

P_1 = absolute initial pressure of the gas, lb. per sq. in.
P_2 = absolute final pressure of the gas, lb. per sq. in.
f = coefficient of friction
l = length of line, ft.

Let

C_a = thermodynamic constant for air
G = specific gravity of flowing gas, air = 1.0
D = diameter of pipe, ft.
d = diameter of pipe, in.

Then

$$C = \frac{C_a}{G}$$

and

$$m = \frac{D}{4} = \frac{d}{48}$$

Hence

$$u_1 = \left[\frac{gC_aT(P_1^2 - P_2^2)d}{48\,GflP_1^2}\right]^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

If

q = quantity of gas flowing per second, based on absolute pressure and temperature of P_o and T_o

A = area of cross-section of pipe in sq. ft. $= \dfrac{\pi d^2}{4 \times 144}$

Then

$$q = u_1 A \frac{P_1 T_o}{P_o T} = u_1 \frac{\pi d^2}{4 \times 144} \frac{T_o}{P_o} \frac{P_1}{T} = \frac{\pi}{576} \frac{T_o}{P_o} \left[\frac{gC_a(P_1^2 - P_2^2)d^5}{48\,GTfl}\right]^{\frac{1}{2}} \ldots [8]$$

If

Q = flow, cu. ft. per hr., based on P_o and T_o
L = length of line, miles

Then

$$l = 5280L$$

and

$$Q = 3600q$$

$$Q = \frac{3600\pi}{576\sqrt{48 \times 5280}} \frac{T_o}{P_o} \sqrt{gC_a} \left[\frac{(P_1^2 - P_2^2)d^5}{GTfL}\right]^{\frac{1}{2}} \quad \ldots\ldots\ldots [9]$$

Taking $g = 32.17$ and $C_a = 53.33$

$$Q = 1.6156 \frac{T_o}{P_o} \left[\frac{(P_1^2 - P_2^2)d^5}{GTfL}\right]^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots [10]$$

37 Experiments on the flow of air in pipes of different diameters indicate that the coefficient of friction f is a variable, decreasing with increasing diameters of line. A great many such experiments have been collected and published in Compressed Air, by Elmo G. Harris, from which, by the use of equation [10], the coefficients of friction have been computed and plotted in Fig. 2.

38 In the reports of these tests no statements were made as to the method of measuring the quantity of gas flowing, and it is quite probable that many of the results are inaccurate in this respect. Notwithstanding this, however, the nature of the variation of f with

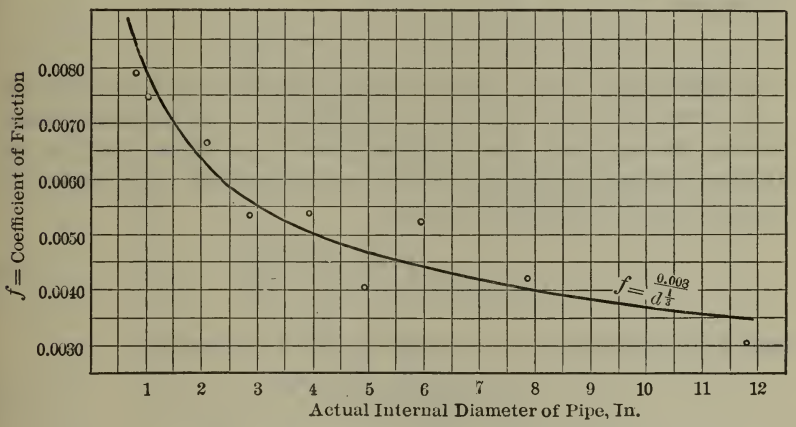

FIG. 2 CURVE SHOWING COEFFICIENT OF FRICTION OF FLOW OF AIR IN PIPES

the diameter is evident, and the curve represented by the equation

$$f = \frac{0.008}{\sqrt[3]{d}}$$

gives a fair average of the loci of the points plotted. Inserting this value of f in equation [10], the expression becomes

$$Q = 18.062 \frac{T_o}{P_o} \left[\frac{(P_1^2 - P_2^2) d^{5\frac{1}{3}}}{GTL} \right]^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots\ldots [11]$$

Equation [11] is the general formula for the flow of gas in long pipe lines.

39 In 1901, Forrest M. Towl conducted an extended test on an 8-in. line, 70 miles long, supplying gas to Buffalo, the results of which were published in a bulletin issued by Columbia University in 1911. Previous to the test the line had been repaired and tested for leaks, and was known to be practically gas tight. The flow was measured

by standardized pitot tubes, which gave results accurate within less than 1 per cent. The specific gravity G of the flowing gas was 0.64, its temperature, 32 deg. fahr., or $T=492$ deg. absolute. The temperature basis on which the gas was measured was 50 deg. fahr., or $T_o=510$ deg. absolute, and the pressure basis was 4 oz. above 14.4 lb., or $P_o=14.65$ lb. per sq. in. absolute. In a length of pipe 70.32 miles long, P_1 and P_2 were 210 and 41 lb. per sq. in. absolute, respectively. The actual diameter of the pipe was 7.981 in., and the rate of flow by pitot tube was found to be 221,000 cu. ft. per hr.

40 Inserting these quantities in formula [6] and solving for flow, it becomes

$$Q=221{,}400 \text{ cu. ft. per hr.}$$

or less than 0.2 of 1 per cent greater than the actual flow as measured.

41 Assuming gas standard conditions of measurement basis, namely, 60 deg. fahr. and 14.65 lb. absolute pressure, and that the average flowing temperature of the gas throughout the year will be 40 deg. fahr., the formula becomes

$$Q=28.66\left[\frac{(P_1^2-P_2^2)d^{5\frac{1}{3}}}{LG}\right]^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [12]$$

and, if an average specific gravity of 0.60 be assumed

$$Q=37\left[\frac{(P_1^2-P_2^2)d^{5\frac{1}{3}}}{L}\right]^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [13]$$

42 Formula [13] is of practical use in designing lines for the transmission of natural gas. It is used as given, or in a transposed form, for all problems relating to single lines of uniform diameter.

43 If a line is composed of several lengths, L_1, L_2, $\ldots L_n$, of diameters d_1, d_2, $\ldots d_n$, each of these lengths must be transformed into an equivalent length of one chosen diameter, by means of the formula

$$L_n^1=L_n\left[\frac{d_n^1}{d_n}\right]^{5\frac{1}{3}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [14]$$

These equivalent lengths added together will give $L=L_1'+L_2'+\ldots+L_n'$, which is the value for L in formula [13].

44 Values of equivalent lengths for different diameters can be most conveniently ascertained by the use of curves in Fig. 3, which consist of plots of the values of $d^{5\frac{1}{3}}$ for varying values of d. Values taken from these curves are convenient to use directly in the pipe-

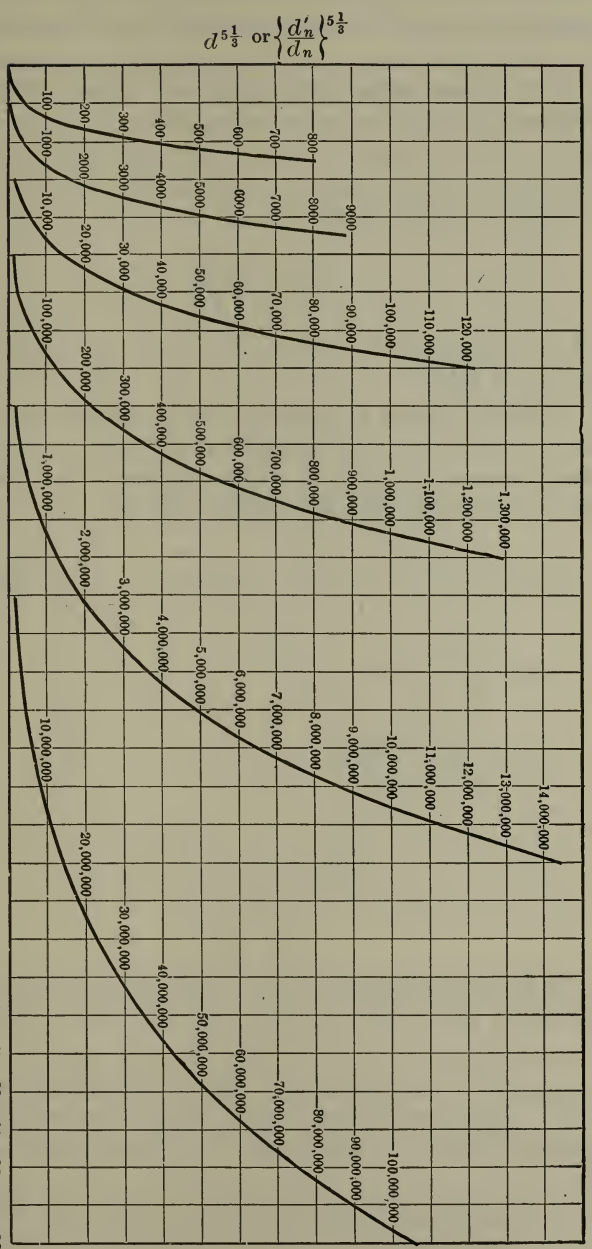

FIG. 3 CURVES SHOWING VALUES OF EQUIVALENT LENGTHS FOR DIFFERENT DIAMETERS OF PIPES

line formula, equation [13], whereas they are most simply used in equation [14] as values of the $5\frac{1}{3}$ power of the diameter ratios.

45 In the next case to be discussed, that of looped systems of lines (two or more pipes paralleling each other), it often happens that the several pipes of a loop have different lengths. It is then necessary, first, to transform all of them into their respective equivalents having a common length. This is done by means of a transposed form of equation [14], namely

$$d'_n = d_n \left[\frac{L'_n}{L_n}\right]^{\frac{3}{16}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [15]$$

It will be seen, of course, that these values can likewise be derived by the use of Fig. 3.

46 Having reduced the pipes to their equivalents of a common length L, let the equivalent diameters be d_1, d_2, $\ldots d_n$. The initial and final pressures P_1 and P_2 will be common to all, and therefore the flow per hour through the system will be

$$Q = 37\sqrt{\frac{P_1^2 - P_2^2}{L}}\left[\sqrt{d_1^{5\frac{1}{3}}} + \sqrt{d_2^{5\frac{1}{3}}} + \ldots + \sqrt{d_n^{5\frac{1}{3}}}\right] \quad \ldots\ldots [16]$$

$$= 37\sqrt{\frac{P_1^2 - P_2^2}{L}}\ \Sigma\ \sqrt{d^{5\frac{1}{3}}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [17]$$

$$= 37\sqrt{\frac{(P_1^2 - P_2^2)d_o^{5\frac{1}{3}}}{L}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [18]$$

where

$$\sqrt{d_o^{5\frac{1}{3}}} = d_o^{\frac{8}{3}} = \Sigma\sqrt{d^{5\frac{1}{3}}} = \Sigma d^{\frac{8}{3}} \quad \ldots\ldots\ldots\ldots\ldots [19]$$

or

$$d_o = \left[\Sigma d^{\frac{8}{3}}\right]^{\frac{3}{8}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [20]$$

47 If the whole line consists of other sections of similarly composed looped lines, the equivalent single diameter of each section can be determined by equation [20] and the section then reduced to an equivalent length of some chosen uniform diameter by equation [14]. These equivalent lengths added together will give a line of uniform diameter of single pipe equivalent to the actual looped system.

48 For facilitating the computation of such a line, Fig. 4 is given, which is a plot of values of $\sqrt{d^{5\frac{1}{3}}}$ for values of d. From these curves

the values of $\sqrt{d^{5\frac{1}{3}}}$ for the several loop diameters are read off and added together, the resulting equivalent diameter d_o being then ascertained from the curve.

49 *Pipe-Line Storage Capacity.* The storage capacity of a pipe line is the excess gas, over and above the average rate of supply, that

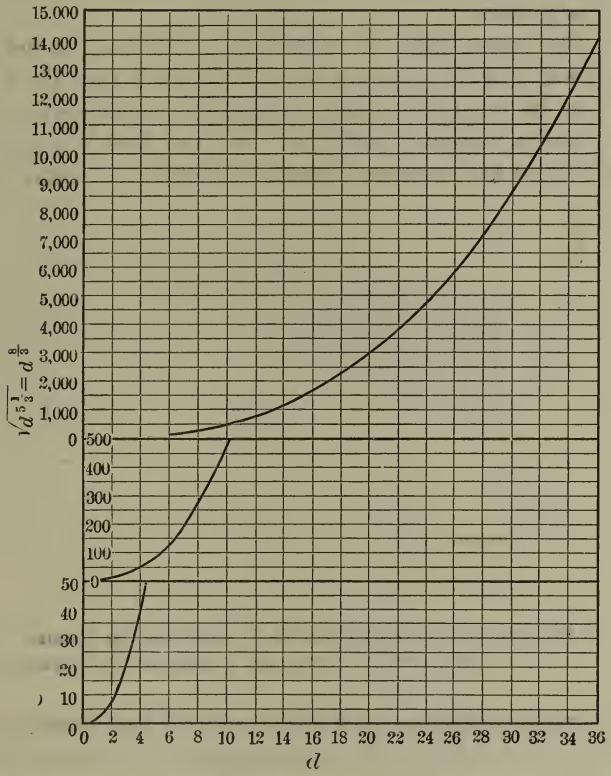

FIG. 4 CURVES GIVING DIAMETER OF PIPE RAISED TO $^8/_3$ POWER FOR USE IN COMPUTING PIPE LINES CONSISTING OF SECTIONS OF LOOPED LINES OF OTHER DIAMETERS

can be packed in the line during the period when the demand is less than the supply. The supply rate is the rate of average daily delivery and since the volume of gas contained in the line is a function of the pressure, it is necessary in order to determine the storage capacity, to develop an expression for the total line contents, which will take into account the variable pressure conditions at all points along the line. Having such an expression, the total contents under the

"packed" and "unpacked" conditions may be computed, and the storage ascertained by taking their difference.

50 The unpacked condition of a pipe line obtains when the gas is flowing at the average daily rate, for when the consumption is below this point all excess gas is being stored, and this point, therefore, is the lower limit, or base of the peaks in the daily delivery curve for the day.

51 The upper limit, or packed line condition, is not as readily determined, but if it is considered to be such that the intake pressure is at its maximum point, as fixed by considerations of safety, station pump pressure limits, etc., and the flow is a mean between the minimum and average rates for the day, the result will be very

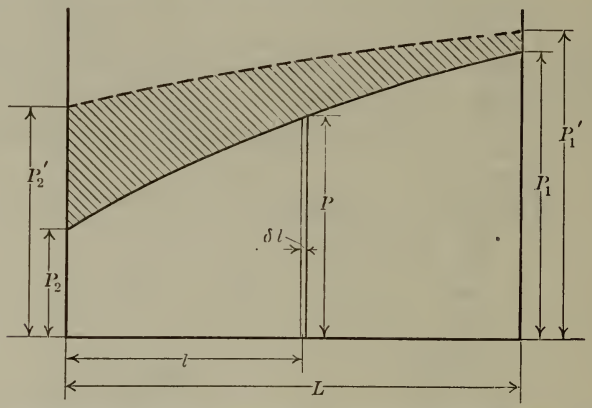

FIG. 5 VARIATION OF INTAKE PRESSURE P_1 FOR VARYING VALUES OF L, LENGTHS OF PIPE, WITH DISCHARGE PRESSURE P_2 CONSTANT

nearly actual conditions, and whatever error may thereby be involved will be on the side of safety in estimating the storage capacity.

52 Fig. 5 shows the variation of intake pressure P_1 for varying values of L with the discharge pressure P_2 constant. Two curves are plotted: The solid line represents the conditions with an assumed average rate of flow and the discharge pressure P_2 fixed by the minimum value allowable by the requirements of the distributing system. The dotted line represents the pressure conditions in the same line with the intake pressure P_1 at the maximum allowable value, and the flow at an assumed mean of the minimum and average rates for the day. This latter curve represents the packed line, and the shaded area the quantity of gas stored in the line available for peak loads.

53 Let P = absolute pressure at any point; then for any given line and flow condition, from equation [13]

$$P^2 - P_2^2 = l\frac{Q^2}{\overline{37}^2 d^5} = Kl \quad \ldots\ldots [21]$$

wherein

$$K = \frac{Q^2}{\overline{37}^2 d^{5\frac{1}{3}}} \quad \ldots\ldots [22]$$

Hence

$$P = \sqrt{Kl + P_2^2} \quad \ldots\ldots [23]$$

The gas contained in a length δl based on any pressure P_o will be in cu. ft.

$$\delta V = 5280\ A \cdot \delta l \cdot \frac{P}{P_o}$$

where A is the cross-sectional area of the pipe in sq. ft. The total quantity of gas in the line under the given conditions will then be

$$V = \int \delta V = 5280 \frac{A}{P_o} \int_o^L \sqrt{Kl + P_2^2}\, \delta l = \frac{3520}{K} \frac{A}{P_o} \left[(KL + P_2^2)^{\frac{3}{2}} - P_2^3 \right] \quad [24]$$

But

$$KL = P_1^2 - P_2^2 \text{ and } K = \frac{P_1^2 - P_2^2}{L}$$

Hence

$$V = \frac{3520\ AL}{P_o(P_1^2 - P_2^2)} \left[P_1^3 - P_2^3 \right]$$

$$= 3520 \frac{AL}{P_o} \left[P_1 + P_2 - \frac{P_1 P_2}{P_1 + P_2} \right] \quad \ldots\ldots [25]$$

$$= 19.20 \frac{d^2 L}{P_o} \left[P_1 + P_2 - \frac{P_1 P_2}{P_1 + P_2} \right] \quad \ldots\ldots [26]$$

54 In these formulae d is the actual internal diameter of the pipe in inches, and L the length of line in miles, the pressures all being lb. per sq. in. absolute.

55 In the case of a complex system it is necessary, first, to ascertain the common pressures at all junction points where the lines are tied together, and then to compute separately the capacity of each section. To do this it is necessary to use the actual diameter, or area, and length of each of the pipes in the loop. The gas content of a looped section thus becomes

$$V=\frac{3520}{P_o}\left[P_1+P_2-\frac{P_1P_2}{P_1+P_2}\right]\Sigma AL \quad \ldots\ldots\ldots\ldots [27]$$

or

$$=\frac{19.20}{P_o}\left[P_1+P_2-\frac{P_1P_2}{P_1+P_2}\right]\Sigma d^2L \quad \ldots\ldots\ldots\ldots [28]$$

56 The gas content of the entire system will thus be the sum of the contents obtained for the several sections of line. If the quantity thus obtained for average flow conditions is deducted from that similarly computed for minimum flow with maximum intake pressure P_1, the result will be the total available storage capacity of the line.

57 If the letters used in equations [27] and [28] be taken to represent the pressure conditions of the packed lines and P_1' and P_2'

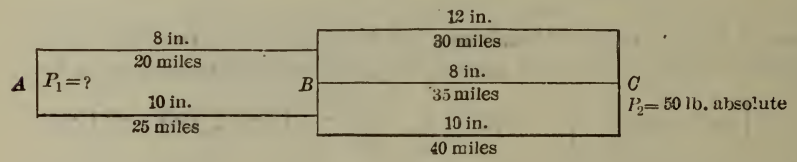

FIG. 5-a DIAGRAM ILLUSTRATING APPLICATION OF PIPE-LINE FORMULAE

those of the unpacked line, the available storage capacity of the system will be

$$S=\frac{3520}{P_o}\left(P_1-P_1'+P_2-P_2'-\frac{P_1P_2}{P_1+P_2}+\frac{P_1'P_2'}{P_1'+P_2'}\right)\Sigma AL \quad \ldots\ldots [29]$$

or

$$=\frac{19.20}{P_o}\left(P_1-P_1'+P_2-P_2'-\frac{P_1P_2}{P_1+P_2}+\frac{P_1'P_2'}{P_1'+P_2'}\right)\Sigma d^2L \quad \ldots\ldots [30]$$

58 The following problem will illustrate the application of all of the above pipe-line formulae. Assume a complex line made up as shown in Fig. 5-a, and let it be required to find the equivalent line of single pipe and the intake, or initial pressure P_1 necessary to force 1,000,000 cu. ft. per hr. through the system, with a terminal pressure $P_2=50$ lb. absolute. Taking first the section from A to B, we have 20 miles of 8-in. (net) pipe and 25 miles of 10-in. By equation [15] the diameter of a pipe 25 miles long that will be equivalent to the 20 miles of 8-in. pipe is, in inches

$$d_n'=8\left(\frac{25}{20}\right)^{\frac{2}{16}}=8.342$$

Section AB is then equivalent to two parallel or looped lines each 25 miles long and having respective diameters of 8.342 in. and 10 in.

$$\begin{array}{ll} \text{For } d = 8.342 \text{ in.} & d^{\frac{8}{3}} = 286.2 \text{ in.} \\ \text{for } d = 10.0 & d^{\frac{8}{3}} = 464.2 \\ & \hline \\ & \Sigma d^{\frac{8}{3}} = 750.4 \text{ in.} \end{array}$$

and by equation [20]

$$d_o = \left(\Sigma d^{\frac{8}{3}}\right)^{\frac{3}{8}} = (750.4)^{\frac{3}{8}} = 11.97$$

or, section AB is equivalent to one pipe 11.97 in. in diameter and 25 miles long.

59 By equation [15] in section BC, the 8-in. pipe, 35 miles long is equivalent to a 7.772-in. pipe 30 miles long, since

$$d'_n = 8\left[\frac{30}{35}\right]^{\frac{3}{16}} = 7.772$$

60 In the same manner, the 10-in. line 40 miles long, is equivalent to a 9.475-in. line 30 miles long. Then, by equation [20]

$$\begin{array}{ll} \text{for } d = 12.0 & d^{\frac{8}{3}} = 754.8 \\ d = 7.772 & d^{\frac{8}{3}} = 237.0 \\ d = 9.475 & d^{\frac{8}{3}} = 402.0 \\ & \hline \\ & \Sigma d^{\frac{8}{3}} = 1393.8 \end{array}$$

$$d_o = \left(\Sigma d^{\frac{8}{3}}\right)^{\frac{3}{8}} = (1393.8)^{\frac{3}{8}} = 15.10 \text{ in.}$$

Hence section BC is equivalent to one pipe 15.10 in. in diameter, 30 miles long. By equation [14]

$$L'_n = 30\left(\frac{11.97}{15.10}\right)^{5\frac{1}{3}} = 8.68 \text{ miles}$$

61 Thus, section BC is equivalent to a single line 11.97 in. in diameter and 8.68 miles long and the whole system is equivalent to a single line 11.97 in. in diameter and 33.68 miles long.

62 For ascertaining the required initial pressure for a flow of 1,000,000 cu. ft. per hr. the known line constants are therefore, $P_2 = 50$, $d = 11.97$, $L = 33.68$, $Q = 1{,}000{,}000$. By equation [13]

$$1{,}000{,}000 = 37\sqrt{\frac{P_1^2 - \overline{50}^2}{33.68}\,\overline{11.97}^{5\frac{1}{3}}}$$

$$P_1 = 215.1 \text{ lb. absolute}$$

63 The relative quantities of gas passed by the several pipes of the loops are to each other as the 8/3d power of their equivalent diam-

eters for equal lengths. The pressure at junction B can be ascertained by means of equation [13], using for L and d the equivalent values derived above for section AB. This pressure is required to be known in order to determine the storage capacity of the line. After having ascertained the above quantities, the pressure drops being known, the computation can be checked by computing the flow through the several lines as they actually exist.

64 Instead of computing the equivalent lengths and diameters by equations [14], [15], and [20], they may be ascertained from the curves of Figs. 3 and 4.

65 To compute the storage capacity of the system, first consider section AB. Assume the maximum allowable pressure at A to be $P_1 = 225$ lb. absolute, and the flow $Q = 800{,}000$ cu. ft. per hr. This section was shown to be equivalent to a single line 11.97 in. in diameter and 25 miles long. Substituting these known values in equation [13], the pressure at B is found to be $P_2 = 172.7$ lb.

66 When the line is unpacked, that is, under average flow conditions of 1,000,000 cu. ft. per hr., P_1' was shown to be 215.1 lb. and by formula [13], P_2' is found to be 117.4 lb.

67 Substituting these values in equation [30], together with the actual dimensions of the pipe in the section of line under consideration, the available storage capacity of this section AB in cu. ft. is

$$S = \frac{19.20}{14.65}\left(225 - 215.1 + 172.7 - 117.4 - \frac{225 \times 172.7}{225 + 172.7} + \frac{215.1 \times 117.4}{215.1 + 117.4}\right)(8^2 \times 20 + \overline{10}^2 \times 25) = 215{,}000$$

68 In like manner the storage capacity of BC is computed, the sum of the two results being the total storage capacity of the whole system available for peak demands.

69 *Power Required to Compress Natural Gas.* Before discussing the general problem of the design of a transmission system, it is necessary to give a rational formula for the power required to pump natural gas. It has generally been assumed that the compression is approximately adiabatic, and the theoretical adiabatic formula has been used for determining the power requirements. Many tests on compressors of widely differing types have convinced the author that this formula sometimes leads to serious errors, and, accordingly, an empirical formula is presented, which is believed to represent the actual operating conditions much more closely than the adiabatic

formula. First, however, both the isothermal and adiabatic formulae will be presented for purposes of comparison.

70 If gas could be compressed isothermally, the theoretical work required, as derived from the laws of perfect gases and with no clearance in the compressor, would be

$$W = P_1 V_1 \log_e \frac{P_2}{P_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots [31]$$

where

W = ft-lb. of work per lb. of gas compressed
P_1 = absolute suction pressure, lb. per sq. ft.
P_2 = absolute discharge pressure, lb. per sq. ft.
V_1 = volume of 1 lb. of gas at pressure P_1, cu. ft.

Let

P_o = absolute pressure, lb. per sq. ft. on which the measurement of the gas is based
T_o = absolute temperature basis of measurement, deg. fahr.
V_o = volume of 1 lb. of gas at P_o and T_o
T_1 = absolute temperature at which the gas is compressed

Then

$$\frac{P_1 V_1}{T_1} = \frac{P_o V_o}{T_o}$$

and

$$P_1 V_1 = P_o V_o \frac{T_1}{T_o}$$

hence

$$W = P_o V_o \frac{T_1}{T_o} \log_e \frac{P_2}{P_1} \text{ ft-lb} \quad \ldots\ldots\ldots\ldots\ldots\ldots [32]$$

71 At P_o and T_o 1,000,000 cu. ft. of gas will weigh $\frac{1{,}000{,}000}{V_o}$ lb, and if 1,000,000 cu. ft. are compressed in one day of 24 hours (1440 minutes), there will be $\frac{1{,}000{,}000}{1440\ V_o}$ lb. compressed per minute. Hence it will require

$$\frac{1{,}000{,}000}{1440\ V_o} \frac{P_o V_o T_1}{T_o} \log_e \frac{P_2}{P_1} \text{ ft-lb. per min.}$$

or

$$\frac{1{,}000{,}000}{1440 \times 33000} \frac{T_1}{T_o} P_o \log_e \frac{P_2}{P_1} \text{ h.p.}$$

72 If p_o = the base pressure in lb. per sq. in. absolute, and p_2 and p_1 are absolute pressures per sq. in. corresponding to P_2 and P_1, $P_o = 144\ p_o$ and

$$\text{h.p.}_i = \frac{1{,}000{,}000 \times 2.3026 \times 144}{1440 \times 33000} p_o \frac{T_1}{T_o} \log_{10} \frac{p_2}{p_1} = 6.978\ p_o \frac{T_1}{T_o} \log \frac{p_2}{p_1} \quad [33]$$

H.p.$_i$ is the theoretical isothermal horsepower per 1,000,000 cu. ft. of gas per day.

73 This formula gives the least possible power required to compress gas at the rate of 1,000,000 cu. ft. per day, and is therefore used as a measure of the efficiency of compression. In other words, "compression efficiency" is the figure obtained by dividing the horsepower as determined from equation [33] by that actually required on the compressor piston.

74 If the compression were performed strictly according to the adiabatic law where n is the ratio of specific heats of the gas at constant pressure and constant volume, the work, neglecting clearance, required for 1 lb. of gas would be, theoretically

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad [34]$$

But since

$$P_1 V_1 = P_o V_o \frac{T_1}{T_o}$$

$$W = \frac{n}{n-1} P_o V_o \frac{T_1}{T_o} \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad [35]$$

75 In order to compress 1,000,000 cu. ft. per day, it would require, as before

$$\text{h.p.}_a = \frac{1{,}000{,}000}{1440 \times 33000 \times V_o} \frac{n}{n-1} 144\ p_o V_o \frac{T_1}{T_o} \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= 3.03\ p_o \frac{T_1}{T_o} \frac{n}{n-1} \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \text{h.p.} \quad [36]$$

76 As already stated, the bases generally adopted throughout the gas country are $p_o = 14.65$ lb. per sq. in. and $T_o = 60 + 460 = 520$ deg.; also, for average natural gas, $n = 1.266$. Substituting these values, and assuming $T_1 = T_o = 520$ deg., formula [33] for isothermal compression, becomes

$$\text{h.p.}_i = 102.23 \log \frac{p_2}{p_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots [37]$$

and formula [36] for adiabatic compression, becomes

$$\text{h.p.}_a = 211.27 \left[\left(\frac{p_2}{p_1} \right)^{0.21} - 1 \right] \quad \ldots\ldots\ldots\ldots\ldots\ldots [38]$$

77 Equations [37] and [38] give the theoretical h.p. per 1,000,000 cu. ft. per day, for isothermal and adiabatic compression respectively, on the assumption that the compressors have no clearance and that there are no losses of any kind. Obviously these conditions are impossible of realization in practice and experience shows that while one would expect the true formula to be somewhere between those for adiabatic and isothermal conditions, due to jacket cooling, in reality this is not the case, on account of the effects of mechanical clearance and slippage losses, to be explained later.

78 For isothermal compression the mean effective pressure in lb. per sq. in., as derived from equation [31], is

$$\text{m.e.p.}_i = 2.3026\ p_1 \log \frac{p_2}{p_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots [39]$$

79 For adiabatic compression, derived from equation [34], it is

$$\text{m.e.p.}_a = \frac{n}{n-1}\ p_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad \ldots\ldots\ldots\ldots\ldots\ldots [40]$$

which, for natural gas, when $n = 1.266$, is

$$\text{m.e.p.}_a = 4.76\ p_1 \left[\left(\frac{p_2}{p_1} \right)^{0.21} - 1 \right] \quad \ldots\ldots\ldots\ldots\ldots\ldots [41]$$

80 If p_2 in equations [39] and [41] be regarded as constant and equal to 100 per cent, and p_1 be allowed to vary from 0 to 100, and the results plotted with values of m.e.p. as ordinates and values of p_1, as abscissae, the resulting curves will be shown as I and A respectively, of Fig. 6. The maximum, or peak load point for the isothermal curve I occurs at

$$p_1 = 0.3679\ p_2, \text{ or } \frac{p_2}{p_1} = e = 2.718$$

For the adiabatic curve A, it occurs at

$$p_1 = 0.3254\ p_2, \text{ or } \frac{p_2}{p_1} = n^{\frac{n}{n-1}} = 3.073$$

81 By referring to equations [37] and [38], it is seen that the power required to pump a given volume of gas is dependent solely upon the ratio of discharge to suction pressures, irrespective of the actual

value of either of them, whereas equations [39] and [41] show the mean effective pressure to depend upon the absolute value of the suction pressure, as well as upon the number of compressions. They show further that, as the suction increases from zero to the value of the discharge pressure, the latter remaining fixed, the mean effective pressure increases, theoretically, from zero to a maximum value, and then decreases to a theoretical zero when the suction is equal to the discharge. As a matter of fact, however, the indicated mean effective pressure in the compressor cylinder will not fall to zero with suction and discharge pressures equal, due to the wire-drawing of the dense gas through the valves, resulting in a diminution during the admission stroke, of the pressure in the cylinder below that in the suction line, and in a piling up, during the discharge stroke, of the pressure above that in the discharge line. Consequently, with suction and discharge pressures equal, the actual indicated power will always be greater than that of either the isothermal or the adiabatic.

82 With $p_1 = 0$, that is, when no gas is admitted to the compressor, the indicator card will show the compression and re-expansion lines to be practically superimposed if the valves are tight, and consequently the condition of zero work with zero suction pressure is very nearly attained in practice.

83 It has been the custom of most engineers to consider that the actual mean effective pressure of compression follows the adiabatic law, and they have therefore used equations [39] and [41], with constants depending upon assumed values of inefficiency, or lost work, according to the judgment and experience of the engineer. Repeated tests have shown, however, that the work curve has characteristics more nearly approaching those of the isothermal than those of the adiabatic, and in every case of which the author has any record the actual curve begins at zero, for zero suction, and rises between the isothermal and adiabatic curves until a point is reached, at about 3.2 compressions, where it crosses the adiabatic and thereafter continues above it. The peak load is found at about $p_1 = 0.42\, p_2$ or $\frac{p_2}{p_1} = 2.38$ compressions.

84 In Fig. 6 are plotted the results of two independent tests on compressors of large size. The circled dots represent the observations of a test reported by James S. Posgate, of the Kansas Natural Gas Company, in a paper[1] read before the Natural Gas Association

[1] Operation of Compression Stations, The Natural Gas Journal, August 1911, p. 20.

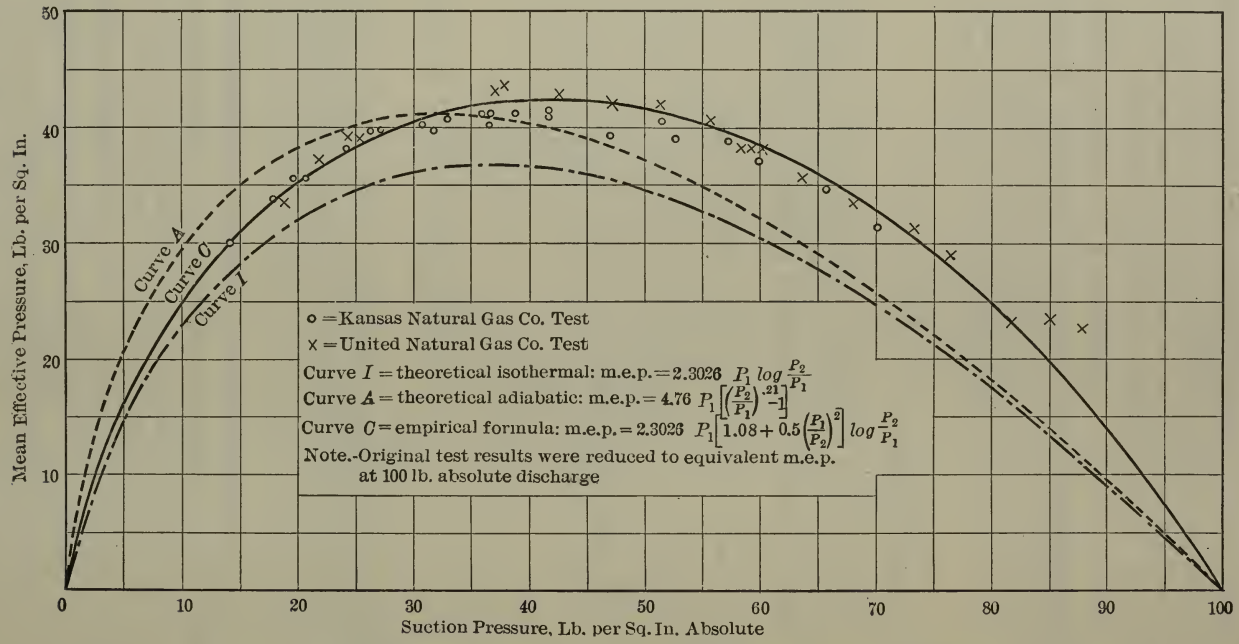

FIG. 6 RESULTS OF TWO INDEPENDENT TESTS FOR COMPRESSORS OF LARGE SIZE

of America at its sixth annual meeting, May 1911. The unit comprised two 17 in. by 48 in. compressor cylinders, operating at 260 lb. gage. The crossed points represent the observations of a test conducted by the author, in March 1911, on a 24¾ in. by 48 in. compressor running at 88 r.p.m. In the latter test the discharge pressure was held constant at about 103 lb. gage, and the suction pressure varied from 88 to 19 per cent of the discharge. The results of the observations in both tests were reduced to equivalent values corresponding to a constant discharge pressure of 100 lb. per sq. in. absolute.

85 A study of the plotted observations of these tests in comparison with the curves *A* and *I* in the same figure, reveals the fact that the actual curve attained in practice resembles the isothermal more

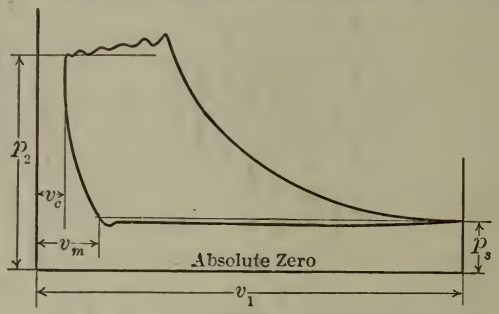

FIG. 7 TYPICAL COMPRESSOR DIAGRAM—SEE PAR. 89

nearly than it does the adiabatic; and that it diverges from the isothermal at a comparatively uniform rate as the suction pressure increases, and can be fairly represented by the equation

$$\text{m.e.p.} = 2.3p_1\left[1.08+0.5\left(\frac{p_1}{p_2}\right)^2\right]\log\frac{p_2}{p_1} \quad \ldots\ldots\ldots\ldots [42]$$

which is plotted as curve *C* in Fig. 6.

86 Curve *C* and its corresponding equation [42] may be taken as fairly representing the conditions met with in practice and has been found by the author to be safe to use in predetermining the indicated horsepower of a compressor.

87 The peculiar relationship existing between the actual and the isothermal curves is doubtless to be explained by the increased rate of cooling of the gas with increasing range of compressions, for the lower the suction pressure, and therefore the greater the number of compressions, the higher will be the temperature of compression and

the less the weight of gas pumped, with correspondingly less total quantity of heat generated. These results combined will therefore cause an increased rate of flow of heat into the cooling water, while at the same time less total quantity of heat will have to be extracted from the gas, with the result that the compression will more and more nearly approach isothermal conditions.

88 Inasmuch as the formula for the mean effective pressure represented by equation [42] is derived from actual operating conditions, it may be considered as taking into account all power losses. In the development of a working formula for the actual power required to pump a given quantity of gas, however, there are a number of sources of loss in volumetric efficiency which must be considered, the most important of which are as follows:

a Mechanical clearance
b Wire-drawing of the gas passing through suction valves and passages
c Heating of the inlet gas while entering the cylinder
d Valve and piston leakage

89 From a study of a great variety of indicator cards taken from many different types of compressors, the author has found that the re-expansion curve closely resembles the isothermal. In fact, many instances were found where its exponent was less than 1.0, doubtless due to discharge valve leakage. Assuming, therefore, that for practical purposes it may be considered as following the isothermal law

$$p_2 v_c = p_s v_m$$

where p_2, p_s, v_c, and v_m are as represented in Fig. 7. Hence

$$v_m = \frac{p_2}{p_s} v_c$$

Volumetric efficiency equals

$$E_i = \frac{v_1 - v_m}{v_1 - v_c} = \frac{v_1 - \frac{p_2}{p_s} v_c}{v_1 - v_c}$$

If the piston displacement = 1.0, then $v_1 - v_c = 1.0$ and $\frac{v_c}{v_1 - v_c} = v_c = m =$ clearance, expressed decimally. Hence

$$v_1 = 1 + m$$

and volumetric efficiency equals

$$E_i = 1 - m\left(\frac{p_2}{p_s} - 1\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots [43]$$

90 Equation [43] expresses the indicated volumetric efficiency of the compressor, i.e., the efficiency as measured on the indicator card.

91 The *indicated output* includes the effect of wire-drawing, or reduction of effective suction pressure of the gas due to passing through the suction valves and passages, as well as that of the indicated volumetric efficiency. In other words, the indicated output is determined by multiplying the piston displacement by the indicated volumetric efficiency and the absolute suction pressure *in the cylinder*, as measured from the cards, and dividing by the absolute pressure base on which the gas is measured. This wire-drawing effect reduces the compressor capacity and increases the power required to pump gas from a given line suction pressure in proportion to the extent of the pressure reduction. This is expressed as a percentage of the absolute line suction pressure. It must be included in the expression for indicated volumetric efficiency by making p_s in equation [43] equal to kp_1, where k is 1.0 minus the percentage drop in suction pressure due to wire-drawing, and p_1 is the line suction pressure, absolute. Hence, the indicated efficiency, expressed in terms of the line suction pressure, is

$$E_i = 1 - m\left(\frac{p_2}{kp_1} - 1\right) \quad [44]$$

92 The third source of loss, heating of the inlet gas entering the cylinder, is an effect that is difficult to determine directly. In 1909 the author conducted a series of tests on six different compressing units, three high-stage and three low, in an effort to discover to what extent this heating occurred, and found that, as was to be expected, it increased with increasing compression ratios, and was affected in a marked degree by any leakage that occurred through discharge valves or past the pistons. Its effect on volumetric efficiency was to reduce it in direct proportion to the ratio of absolute temperature in the suction line to that of the cylinder full of gas at the beginning of compression, and amounted roughly to about 1 per cent for each 5 deg. fahr. difference.

93 The fourth factor, namely, the loss due to leakage, reduced both the volumetric and the work efficiencies. While leaky suction valves caused a flattening of the compression line of the card and thus a reduction in the mean effective pressure, the quantity of gas pumped was also reduced with the net result of an increase in the power required for a given actual quantity of gas discharged. On the other hand, a leak back through the discharge valves into the

cylinder reduced the quantity of gas drawn in through the suction, not only by displacement, but also by its heating effect. It also raised the pressure during the compression stroke, making the compression line steeper, and thereby caused an increase in the mean effective pressure. The ultimate result, therefore, was an increased power requirement to pump a given quantity of gas.

94 In accounting for the losses due to heating and to leakage, they are usually combined with all unaccounted for losses, and called "slippage," for which a certain fixed percentage is assumed and inserted in the expression for volumetric efficiency.

95 Let s, expressed as a decimal, be the slippage effect. Then the true volumetric efficiency, or ratio of gas actually pumped to that computed from piston displacement and line suction pressure p_1, is

$$E = k\left[1 - m\left(\frac{p_2}{kp_1} - 1\right) - s\right] \quad [45]$$

and the quantity of gas pumped in one day of 1440 minutes would be, in millions of cubic feet, based on T_o and p_o

$$Q_m = \frac{1440}{1{,}000{,}000}\,\frac{A}{144}\,LN\,\frac{T_o}{T_1}\,\frac{kp_1}{p_o}\left[1 - m\left(\frac{p_2}{kp_1} - 1\right) - s\right] \quad [46]$$

The horsepower required on the compressor piston will be

$$\text{h.p.} = \frac{\text{m.e.p.} \times L \times A \times N}{33000}$$

The horsepower per 1,000,000 cu. ft. per day will thus be equal to

$$\text{h.p.}_m = \frac{\text{h.p.}}{Q_m} = \frac{100}{33}\,\frac{T_1}{T_o}\,\frac{\text{m.e.p.}}{\dfrac{kp_1}{p_o}\left[1 - m\left(\dfrac{p_2}{kp_1} - 1\right) - s\right]} \quad [47]$$

Substituting the value of m.e.p. as given in equation [42]

$$\text{h.p.}_m = \frac{100}{33} \times 2.3026 \times p_o \times \frac{T_1}{T_o} \times \frac{1.08 \times 0.5\left(\dfrac{p_1}{p_2}\right)^2}{k\left[1 - m\left(\dfrac{p_2}{kp_1} - 1\right) - s\right]} \log \frac{p_2}{p_1} \quad [48]$$

or, if $\dfrac{p_2}{p_1} = r$

$$\text{h.p.}_m = 6.978\, p_o\, \frac{T_1}{T_o}\, \frac{1.08 + \dfrac{1}{2r^2}}{k\left[1 - m\left(\dfrac{r}{k} - 1\right) - s\right]} \log r \quad [49]$$

96 For average conditions in practice, $m=0.02$, $k=0.98$, $s=0.04$, and $p_o=14.65$. Substituting these values in [49]

$$\text{h.p.}_m=102.2\ \frac{1.08+\frac{1}{2\,r^2}}{0.96-0.02\,r}\log r \qquad [50]$$

97 The efficiency of compression is equation [37] divided by equation [50], or

$$E_o=\frac{0.96-0.02\,r}{1.08+\frac{1}{2\,r^2}} \qquad [51]$$

98 In two-stage compression, for equal work in both stages, the power required is equal to twice that necessary to pump the gas through a single stage in which the number of compressions is equal to the square root of the total two-stage compression range. Therefore, if h.p.$_2$= the horsepower per 1,000,000 for two-stage operation

$$\text{h.p.}_2=2\times 6.978\ p_o\ \frac{1.08+\frac{1}{2r}}{k\left[1-m\left(\frac{\sqrt{r}}{k}-1\right)-s\right]}\log\sqrt{r}$$

$$=6.978\ p_o\ \frac{1.08+\frac{1}{2\,r}}{k\left[1-m\left(\frac{\sqrt{r}}{k}-1\right)-s\right]}\log r \qquad [52]$$

Substituting the same constants as were assumed for equation [50]

$$\text{h.p.}_2=102.2\ \frac{1.08+\frac{1}{2\,r}}{0.96-0.02\sqrt{r}}\log r \qquad [53]$$

99 The simplest method of using the horsepower formula is by means of a plot giving values of horsepower per 1,000,000 cu. ft. of gas per day corresponding to values of $\frac{p_2}{p_1}=r$. Equations [37], [38] and [50] are so plotted in Fig. 8, together with the efficiency of compression, equation [51].

100 It will be seen that whereas the actual mean effective pressure in the compression of natural gas is less than that of true adiabatic compression for ranges above 3.2 compressions, the actual horsepower required to pump a given quantity of gas is always

greater, no matter what the compression range may be. This is due to the effect of clearance and the other losses discussed above.

101 The curves in Fig. 8 give the indicated horsepower required on the compressor piston, and in order to determine that necessary in the power cylinders, the values taken from the curves must be increased according to the mechanical efficiency of the machine.

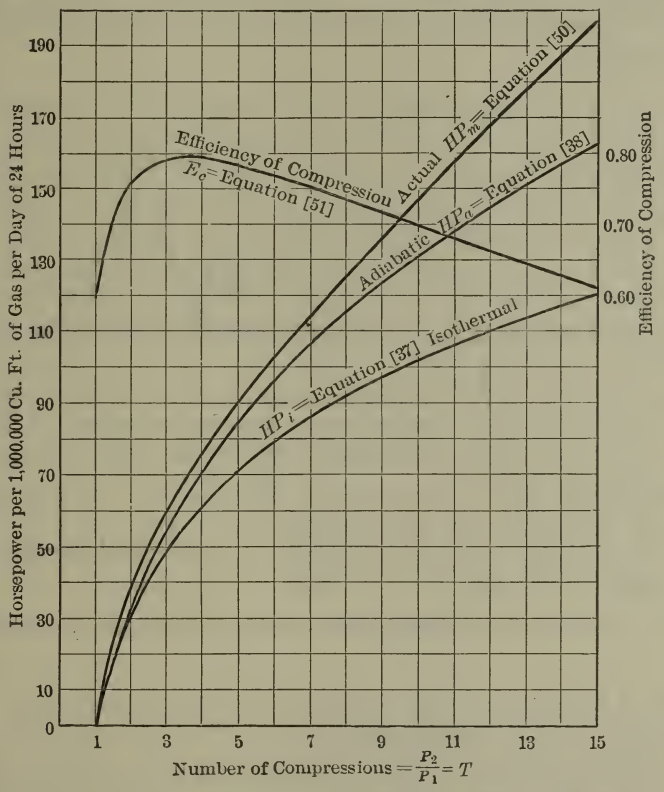

FIG. 8 PLOT OF EQUATIONS [37], [38], [50] AND [51]

This efficiency will range from 75 to 80 per cent in direct-connected gas-engine-driven compressors of large size.

102 Single-stage compression is advisable for ranges up to seven or eight compressions for the reason that the benefit that would be derived from two-stage operation is overbalanced by the excess first cost of the units, combined with the greater cost of operating, installing, and housing them. As the compression range increases above this point, however, two-stage operation becomes necessary

from an operating standpoint, and its advantage from an economical standpoint becomes more and more pronounced.

103 In compressing natural gas through two or more stages it is found to be impracticable to drive the compressors of all stages by the same power unit, on account of the variable pressure conditions which unbalance the loads of the different stages except at one particular ratio of compression. For this reason it is the usual practice to have the compressors of each stage driven by their own individual power units.

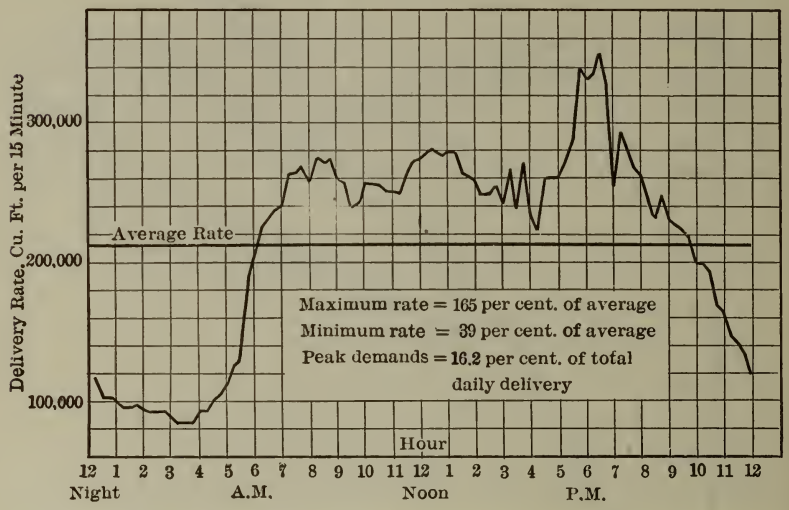

FIG. 9 LOAD CURVE SHOWING DELIVERY OF GAS DURING A COLD DAY IN WINTER

104 It is seldom, if ever, advisable to pump through more than two stages in natural gas transmission, for the reason that the pressure is limited by the strength of the pipe lines to a value that would not warrant the added complication and cost of multi-stage units higher than two.

105 *Pipe-Line Design.* In order to design a transmission system it is necessary to consider the character of service to be rendered and the various load factors to be expected. For domestic service in large cities the maximum daily delivery during the year will be about twice the daily average for the year, while the minimum will range from 25 to 30 per cent of the average. Thus, if 7,300,000,000 cu. ft. of gas are to be delivered in a year, or an average of 20,000,000 per day, provision will have to be made to transport 40,000,000 dur-

ing the coldest day of winter. The load curve for a typical cold day must then be considered and the pipe lines and stations designed not only to deliver the total quantity for the day at the average hourly rate, but also to provide sufficient storage capacity to carry the load safely over the peaks. A typical load curve for a cold day is shown in Fig. 9, and it will be seen that there are three main peaks, each occurring about meal time. As a rule, the demand for gas during the peaks will amount to about 16 per cent of the total daily delivery. The line storage capacity will therefore have to be equal, at least, to this amount. It is thus necessary to provide a line of sufficient capacity to transport the daily average with ease, and to have the requisite storage capacity. At the same time the pumping stations will be required not only to pump the average quantity of gas demanded, but also to pump it to a pressure sufficiently high to pack the line, and thus give it the proper storage. This packing must, of course, be done during periods of depression in the daily load curve, when the demand is less than the supply from the stations.

106 There are many variables involved in the proper design of a pipe line for greatest economy, such as: the pressure to be adopted; the size of the pipe with its limiting safe working strength; the number and location of pumping stations, and whether single or double stage, etc. All depend upon the conditions applying to the particular problem in hand. The fundamental principles upon which a line must be planned are those expressed by the pipe line and horsepower formulae already given, a fact which seems to the author to justify the space devoted to them.

107 In order to secure the greatest economy in design it is necessary to study the effects of the various factors involved in any given case, for there exists a certain relationship among pressures, pipe diameters and number of pumping stations that will give the maximum of economy for any given output and distance of transmission.

108 As a general rule, it is desirable to design the system for the maximum safe working pressures that the pipe will stand, for the reason that high pressures demand smaller pipes, and usually the pipe line is the heaviest item of cost of a system. High pressures, however, mean more power necessary in the compressor stations, with correspondingly increased cost of pumping the gas, thus partially, or perhaps completely, counteracting the lower cost of the smaller lines due to such high pressures. Line leakage is also much more serious the higher the pressure, and on this account it is neces-

sary to make the safe working pressure limits of the pipe line much lower than would be indicated by testing a single joint of the pipe alone.

109 The quantity of gas to be delivered also has a marked influence on the economical number of stations to be used, for the power necessary to pump it is directly proportional to the amount of gas pumped, and therefore the operating and first costs of stations likewise, whereas the size of the line, under the same pressure conditions, increases as the $\frac{3}{8}$th power of the quantity. Consequently, the greater the quantity, the greater will be the relative cost of power equipment over that of the line.

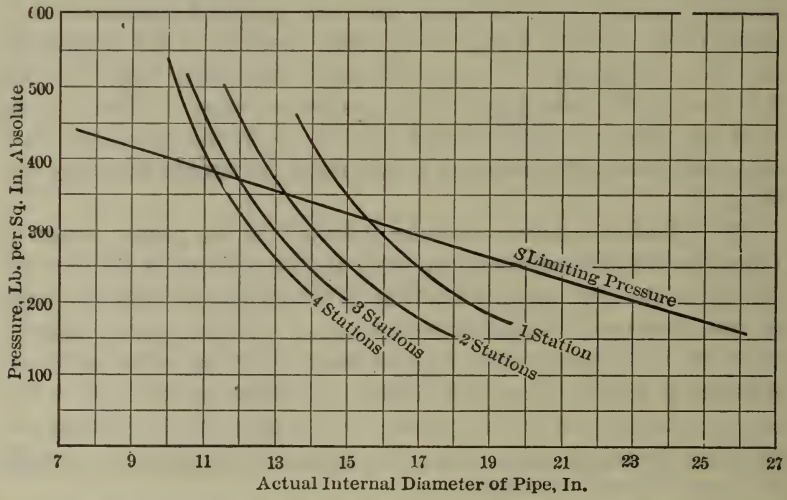

FIG. 10 PIPE DIAMETERS FOR INITIAL LINE PRESSURES FOR INSTALLATIONS OF FROM ONE TO FOUR STATIONS

110 Relay stations between the field and the delivery point should always be single stage only, for it requires the same amount of power to compress the gas through the first stage, say from atmosphere to 53 lb. gage, as it does through the second stage from 53 lb. to 300 lb., whereas practically no benefit is derived in the line for the expansion from 53 lb. to atmosphere, as compared with that from 300 to 53. This is seen from the pressure factor in the pipe-line formula.

111 In order to illustrate the effects on the problem of the various factors involved an example is worked out and plotted. It is assumed that the average daily delivery of a transmission system

during the coldest day is to be 24,000,000 cu. ft., or at a rate of 1,000,000 per hour. The transmission distance is to be 300 miles and it is required to determine the pressures and size of line as well as the number of pumping stations required. For the purposes of the example it is assumed that the total power installed will be equal to that determined by the formula without allowing for reserve capacity, and that the actual pipe diameters are used as derived, without regard to standard sizes. The cost of pipe line is taken as $900 per mile per inch of diameter, and the cost of compressor stations at $100 per compressor horsepower installed. Interest and depreciation on the line are taken at 10 per cent and on the power stations at 12 per cent, while operating costs are figured at $15 per year per installed compressor horsepower. The minimum allowable delivery pressure at the end of the line, where it feeds into the distributing system, is assumed as 50 lb. absolute, while the gas from the field is supposed to enter the compressors at 15 lb. absolute, or about atmosphere.

112 First, assuming one station in the field to pump the gas the whole distance, the required pipe diameters are figured for different initial line pressures by means of the pipe-line formula, equation [13]. These different values of pressure and diameter are then plotted as shown in curve 1, Fig. 10. In this figure the limiting safe pressures for various sizes of pipe line are also shown by the line S, and it is seen that this line and that of the required pressures cross each other. For any diameter less than that indicated by the intersection of line S with curve 1, the pressure would be too high for safe working and this point therefore indicates the minimum diameter and the maximum initial pressure that can be adopted with one station.

113 On the basis of the minimum diameter thus derived, the cost of the line is computed, and from the maximum pressure the power required to pump the gas is determined by means of the horsepower formula, using the proper value of suction pressure. From these computations the yearly cost is determined for a single-station equipment.

114 Then an intermediate station is assumed to be located midway of the distance. In computing the relative pressures and diameters it is first assumed that the intermediate station will pump through a single stage of 3.5 compressions, corresponding to the point of maximum compression efficiency as shown in Fig. 8. Under this assumption the terminal pressure of the first section of line is

$$P_2 = \frac{P_1}{3.5} = 0.2855P_1 \quad \text{and} \quad P_1^2 - P_2^2 = 0.9184P_1^2$$

With this value applied to the pipe-line formula the initial pressure for the field station is computed, assuming it to be the same for both stations. This causes the intermediate discharge pressure to be

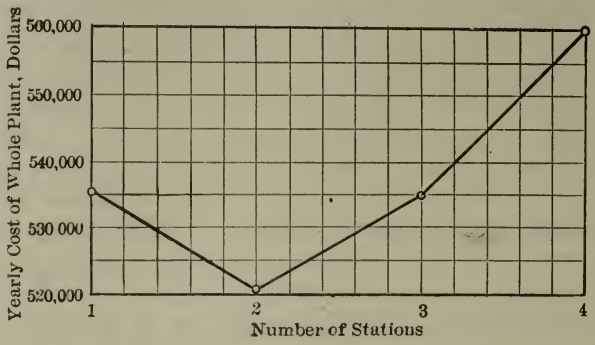

FIG. 11 CURVE SHOWING YEARLY COSTS AGAINST NUMBER OF STATIONS

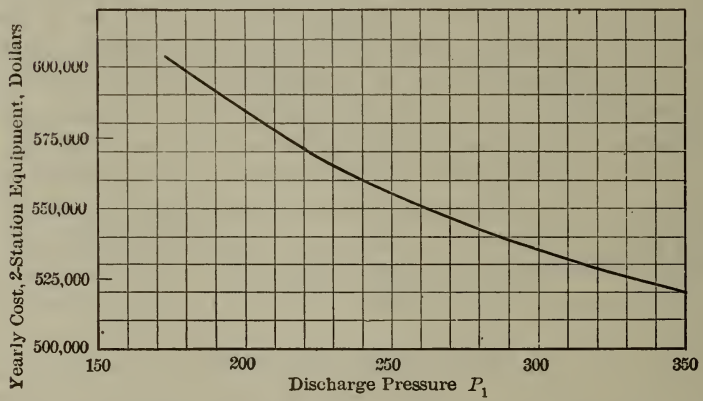

FIG. 12 CURVE SHOWING HIGHEST SAFE PRESSURES GIVING MAXIMUM ECONOMY

slightly higher than would actually be required to transmit the gas through the last section of line with its terminal pressure at 50 lb., but the increase is small, and the method insures the field station discharge pressure being at the proper value.

115 Values of pressure and diameter are plotted and the minimum diameter and the maximum pressure available are taken from the plot as before. The horsepower per 1,000,000 ft. of gas at the intermediate stations will then be constant at 70 h.p., corresponding to

3.5 compressions, but the power of the field station must be figured for its proper ratio of compressions.

116 In the same manner the yearly cost is figured for three and four-station installations, and it is seen from Fig. 11, which is a plot of yearly costs against the number of stations, that for this particular case the lowest cost is attained with but two stations.

117 Figures made with all the conditions the same except that twice the quantity of gas is to be delivered, will show that a one-station equipment furnishes the most economical arrangement. Likewise with smaller maximum quantities to be delivered through the same distance, it can be shown that a greater number of stations, with correspondingly higher pressures and smaller line will result in the maximum economy.

118 Assuming the two-station layout and figuring yearly costs with decreasing initial line pressures, the curve in Fig. 12 results, showing that for this particular case the highest pressures safe to use give the maximum economy.

119 Having thus ascertained the most economical arrangement, it is necessary to consider the question of storage capacity, and see that when the line is packed the pressures and line sizes are adequate to provide sufficient storage gas to take care of the peak load.

120 If this is done for the example chosen it will be found that when the light deliveries are taking place at night the terminal pressure will back up to such an extent that with the intermediate station discharge pressure remaining constant the line will fill with gas to such an extent that more than sufficient storage will exist in the line. This illustrates one distinct advantage of large lines, especially at the terminal end of a system, where the gas can be stored at high pressure near the point of delivery and thus make it possible to draw on the reserve supply on short notice.

STATION DESIGN

121 In many natural gas fields, the natural pressure of the gas is sufficient to carry it to the market. The pressure declines, however, as the gas is withdrawn from the pools, and eventually a point is reached when it is necessary either to build more pipe lines to carry the gas, or to install pump stations for raising the pressure sufficiently high to make the existing lines serve. The principles already set forth will govern the choice in such a case. Usually, however, the pump station will be found preferable, for the reason that the assured continuance of the decline of pressure will eventually

result in the necessity for a station in any event, and, if installed in the beginning, it can readily be adapted to suit changing conditions.

122 Since the volume of gas increases as the pressure diminishes, the size of line necessary to convey a given quantity must increase likewise. Also the pressure drop becomes great under low-pressure conditions. In order, therefore, to make the work of the compressors effective in producing a pull on the wells without excessively low suction pressures at the station, it is necessary to have the stations located as near the center of the field as possible, thus reducing the length of suction lines to a minimum. The addition of a few miles to the length of the discharge line means almost nothing compared with their effect on the suction side. The location of compressor stations, therefore, means not only a study of the conditions relative to water supply and accessibility, but also of probable future developments in the gas production.

123 In passing, it is well to call attention to the necessity for the utmost care in making all suction lines absolutely tight to avoid the possibility of air entering them, and thus endangering the entire system, as well as impairing the service.

124 The type of compressor in general use for natural gas is the ordinary form of reciprocating air compressor. It must be equipped with automatic, or "voluntary," discharge valves, due to the fact that the expulsion of gas from the cylinder begins at differing periods of the stroke, depending on the compression ratio. The most satisfactory suction valve is likewise of this type, although mechanical suction valves have been tried with varying degrees of success. Because of the varying ratios of compression usually encountered, it is advisable to keep the clearance space in compressors as small as possible, for the higher the compression range, the lower will be the volumetric efficiency.

125 The design of the station units involves not only an assumption of the volume of gas to be pumped, but also the determination of the maximum discharge pressure to be demanded of the compressors.

126 The quantity of gas is usually estimated from a consideration of the expected demand for it, as well as a knowledge of the available supply. In most fields there are two general classes of wells, designated as "high pressure" and "low pressure," the former usually being the new ones with natural pressures sufficient to permit them to feed into the lines "ahead" of the pumps, that is, into their discharge lines. The latter, or low-pressure wells, are those that

require pumping. Frequently there are wells with pressures too low for feeding into the high-pressure discharge, but sufficiently high to feed the suction of the high-stage pumps, and by proper systems of piping, this is usually provided for. This naturally affects the station design, as it necessitates the pumping of more gas through the second stage than through the first, and proper provision must be made for it by the addition of the requisite number of high-stage units.

127 No fixed value of discharge or suction pressures obtains in the operation of any natural gas pumping station. The variation in these pressures throughout a single day may cover a wide range of limits, while it is especially marked from day to day throughout the year. In this respect the compression of natural gas differs widely from that of air under the usual conditions, and presents several additional problems not encountered in the compression of air. As related to the question of design, this factor determines not only the power necessary to drive the compressor but also the quantity of free gas a machine will handle, and involves what is termed the peak-load condition of power requirement, as was shown in the development of the power formula. While it is well to provide for ample power to drive the compressors under all conditions of operation, the principle is easily carried to absurd limits, with the ultimate result of great loss in volumetric output. In some instances it is found necessary to pump gas with compression ranges varying from two to five or six. Under these conditions the unit must be designed to carry the load over the peak, and it will then operate under any and all conditions of suction pressure, with the discharge at its proper value. Usually, however, this will not be necessary, and then the upper limit of suction pressure may be estimated, and, if on occasion it should greatly exceed the value for which the units were designed, it can be reduced by throttling the gates on the suction lines.

128 The discharge pressure to be used in designing the compressor cylinders must be that of maximum condition, which is usually at night when the lines are being packed, or stored, with gas to provide sufficient storage capacity to help over the peak load conditions in the distributing system.

129 While this maximum pressure must be used to determine the engine power required, the ratio of cylinder sizes for two-stage compression must conform more nearly to average load conditions, in order properly to equalize the work between the two stages for average conditions of operation. Once the cylinder diameters are fixed, the ratio of discharge to suction pressure of the low-stage unit,

or, in other words, the ratio between the high and low-stage suction pressures, will be practically independent of the high-stage discharge pressure. The ratio of cylinder diameters may be determined as follows:

130 Let

V_l = the volume of low-pressure cylinder per revolution
D_l = diameter of cylinder
d_l = diameter of piston rod
s_l = stroke
n_l = speed, r.p.m.
e_l = volumetric efficiency
P_l = absolute suction pressure
T_l = absolute suction temperature

and let the corresponding letters, with sub h, apply to the high-stage compressor. Also, let P_o and T_o be the absolute pressure and temperature of any base on which the gas may be measured. Then the actual quantity of gas pumped per minute referred to P_o and T_o will be

$$Q = V_l \frac{P_l}{P_o} \frac{T_o}{T_l} e_l n_l = V_h \frac{P_h}{P_o} \frac{T_o}{T_h} e_h n_h \quad \ldots\ldots\ldots\ldots [54]$$

since all the gas that passes through the low-stage cylinder must be pumped through the high. Hence

$$n_l V_l e_l \frac{P_l}{T_l} = n_h V_h e_h \frac{P_h}{T_h}$$

$$\frac{P_h}{P_l} = \frac{V_l}{V_h} \frac{e_l}{e_h} \frac{T_h}{T_l} \frac{n_l}{n_h} \quad \ldots\ldots\ldots\ldots\ldots\ldots [55]$$

But

$$V_l = \frac{\pi s_l}{4} \left(D_l^2 - \frac{d_l^2}{2} \right) = \frac{\pi s_l}{8} (2 D_l^2 - d_l^2)$$

and

$$V_h = \frac{\pi s_h}{8} (2 D_h^2 - d_h^2)$$

Hence

$$\frac{P_h}{P_l} = \frac{s_l}{s_h} \frac{(2 D_l^2 - d_l^2)}{(2 D_h^2 - d_h^2)} \frac{e_l}{e_h} \frac{T_h}{T_l} \frac{n_l}{n_h} \quad \ldots\ldots\ldots\ldots\ldots [56]$$

Assuming the same speed and stroke for both stages

$$\frac{P_h}{P_l} = \frac{(2 D_l^2 - d_l^2)}{(2 D_h^2 - d_h^2)} \frac{e_l}{e_h} \frac{T_h}{T_l} \quad \ldots\ldots\ldots\ldots\ldots [57]$$

131 From equation [57], it will be seen that the ratio of suction pressures in the two stages depends upon the net cylinder areas, the volumetric efficiencies and temperatures of the suction gas.

132 If the intercooling is inefficient, T_h will exceed T_l, and the discharge pressure of the low-stage compressor ($=P_h=$the high-stage suction) will be raised, thus imposing more work on the low-stage units. Usually, however, adequate cooling can be attained,

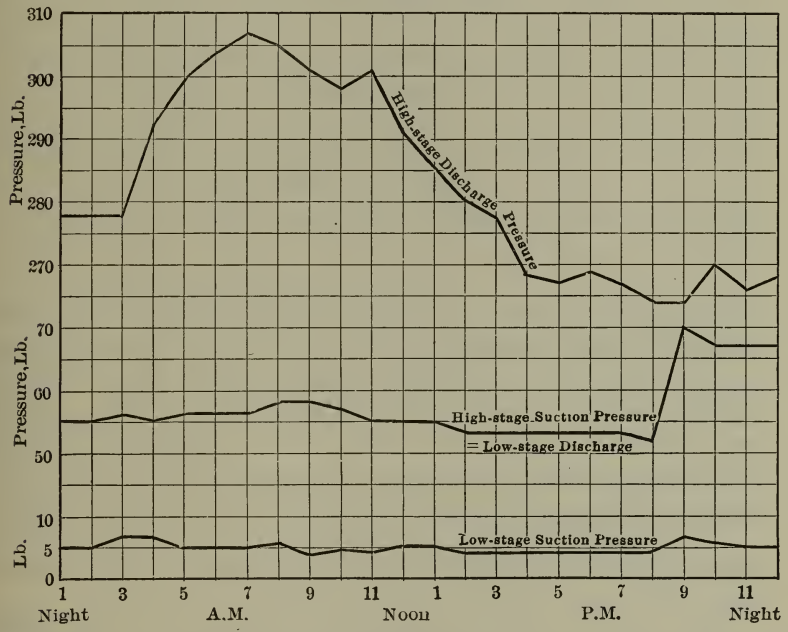

FIG. 13 CURVE SHOWING HOURLY PRESSURES OF TYPICAL PUMPING STATION FEEDING INTO A LINE 120 MILES LONG DURING AN AVERAGE WINTER'S DAY

and the ratios of temperatures may therefore be taken as one. Under this assumption the compression ratio in the low-stage cylinder becomes equal to the ratio of net effective areas of low to high-stage cylinders, that is, the computed areas multiplied by the respective volumetric efficiencies.

133 The ratio of the diameters of the high and low-stage cylinders can therefore be determined from equation [57], having fixed upon the proper ratio of compressions for each stage, as determined by average operating conditions.

134 In order to illustrate the fact previously stated, that the high-stage suction pressure is practically independent of its dis-

charge, Fig. 13 is given, showing a plot of the hourly pressures of a typical pumping station feeding into a line 120 miles long during an average winter's day.

135 The size of compressor cylinder required depends upon three things: the quantity of gas to be pumped, the suction pressure at which it can be supplied to the compressor, and the speed of the compressor.

136 The first of these requirements must be ascertained from the number of units to be installed, and, in this particular, there seems to be a considerable difference of opinion among engineers. The author believes that there should be a sufficient number of units, so that in case of a shutdown of any one of them, the capacity of the plant would not be seriously impaired. In fact, it is distinctly the part of wisdom to provide one spare unit over and above the actual requirement under the most severe conditions. Natural gas compressors, unlike most other power apparatus, cannot have their capacity forced, for there are only two means of increasing it, namely, by increasing the speed, which has a well defined upper limit, and by raising the suction pressure; and usually, when the greatest demand is made on the plant, the field is supplying all the gas of which it is capable, and thus it is impossible to add to this pressure. Natural gas companies are, as a rule, public service corporations, and it is absolutely necessary to maintain the operation of the stations continuously during severe weather in order to prevent great suffering.

137 Having given the quantity of gas to be pumped and the suction pressure, the required size of the cylinder is readily computed when the proper value of volumetric efficiency is assigned.

138 In good modern compressors of large size and long stroke, the indicated volumetric efficiency may range from 98 per cent to 90 per cent, depending upon the ratio of compressions, and the slippage loss will range from 3 per cent to 10 per cent, so that it is not unusual to find actual volumetric efficiencies as high as 94 per cent. In the majority of cases, however, the actual efficiency will range from 80 to 90 per cent, with 85 as a fair average for working conditions.

139 The constantly growing demand for compressors of high efficiency for natural gas compression has given a considerable impetus to improvements in design, which have resulted in lower clearance spaces and improved valves and jacket cooling, and it is entirely reasonable to specify volumetric efficiencies of 90 per cent, or even higher, with units of large size.

140 When the actual volumetric efficiency of a unit is known, its pumping capacity can conveniently be determined from the formula:

$$Q = 0.000909 \times E \times s \times \left(D^2 - \frac{d^2}{2}\right) \frac{NPT_o}{P_o T} \quad \ldots\ldots\ldots [58]$$

where

Q = output, cu. ft. per min.
E = volumetric efficiency of compressor
s = stroke, in.
D = compressor cylinder diameter, in.
d = compressor piston diameter, in.
N = revolutions per minute
P = suction pressure, lb. per sq. in. absolute
T = suction temperature, absolute
P_o = pressure base of gas measurement, lb. per sq. in. absolute
T_o = temperature base, absolute

141 This formula assumes that the compressor has no tail rod. Should it have one, the value in the parenthesis would have to be modified to suit. From the formula the diameter necessary for any given capacity can be computed by substituting the known quantities in the equation.

142 Usually the temperature of the suction gas is assumed to be the same as that of the measurement basis in stating the capacity of a compressor, and hence the factor $\frac{T_o}{T}$ in the formula is dropped, being equal to 1.

GENERAL REMARKS

143 The problem of the number of compressions to use in the determination of the power required is usually more or less indeterminate and requires the exercise of the judgment of the engineer in each particular case, based upon a knowledge of the field pressure conditions and the probable discharge pressure as already discussed. The discharge pressure is, as shown, a matter of computation, but the suction pressure may vary widely under different conditions.

144 Compressors for natural gas were originally driven by steam engines, but, owing to the recent improvements in the design and reliability of large gas engines, and to their high economy, this type of prime mover is rapidly supplanting the steam engine. Usually

the compressor is direct-connected to the engine, although there are many successful installations using the belt drive. Motor-driven units are rare, however.

145 The tendency in recent years has been to lay gas transmission lines with rubber-packed sleeve couplings, in sizes above 4 in. In fact, it is imperative with lines above 10 in. in diameter, owing to the difficulty of making tight screw-collar joints with the larger sizes of pipe. Lines are usually designed for pressures as high as the pipe will stand, sometimes exceeding 400 lb. per sq. in., and, inasmuch as the leakage is a function of the pressure, irrespective of the quantity of gas flowing through the line, it is essential that the pipe joints be made as tight as possible.

146 In using the sleeve type of coupling it must be remembered that it offers very little resistance to end thrust, and proper provision must be made for anchoring the pipe wherever it deviates from a straight line, such as making turns or going over the crest of a knoll.

147 Heavy fittings are usually necessary, not only to stand the gas pressures, but also to provide against the enormous strains produced by expansion and contraction due to temperature changes and to frost. This is especially true at compressor stations because of the high temperatures due to compression, which sometimes reach 300 deg. to 350 deg. fahr. At these stations it is usual to provide swing-joint connections; all stiff joints should be avoided, especially where any considerable length of pipe is involved.

148 Owing to the possibility of leakage gas accumulating in a pump station and endangering the plant, it is necessary to provide adequate ventilation. The method favored by the author is to have the pump house equipped with a monitor, and a fan installed in the auxiliary building, which draws its supply of fresh air from outside, over the hot gas engine exhaust pipes which are laid in a conduit in the pump house. The air is then forced back into the pump house, thus providing both heat and ventilation without the necessity of auxiliary boilers or fires about the premises.

149 All buildings must be lighted by electricity, the generator being installed in the auxiliary building. This generator is usually of the direct-current type in order to provide current for charging the storage batteries generally installed for ignition purposes. The auxiliary building should be from 30 to 40 ft. away from the main pump house in order to lessen the risk of explosion.

150 All electric lamp sockets in the main pump house should be

of the keyless type, and all electric switches should be enclosed. Care should be taken to exclude any pockets under the pump-house floor or elsewhere where gas might collect.

151 In piping up gas compressor cylinders it is strongly advisable to provide a by-pass for use in starting, especially with gas-engine-driven units; and also an automatic safety valve, connected to the discharge pipe between the compressor cylinder and the first gate. There have been several instances where compressors have been destroyed by attempting to start with a closed discharge gate, and this precaution is most necessary. Such relief valves should have ample capacity to pass all the gas the compressor might discharge, and should be tested at regular intervals to insure their being in an operative condition at all times. They are fully as important as a safety valve on a boiler.

152 Two-stage pumping stations should be provided with both inter-coolers and after-coolers, which are usually composed of networks of pipe immersed in a pond or reservoir adjoining the station. Drips should be connected at the discharge end of all coolers, in order to collect and dispose of the gasolene which usually is produced by the process of compressing and cooling natural gas.

153 These drips are sometimes provided on an elaborate scale, where it is found that the gasolene production is sufficiently large to warrant collecting and selling it, and in some cases refining plants are installed for rendering the gasolene fit for the market.

154 It is important to extract all of this gasolene from the lines, as otherwise it will collect in depressions and "freeze," completely plugging the line. The secret of getting out the gasolene is so to build the after-cooler as to cause the flow of gas to be greatly retarded, when the liquid may be extracted by proper separators.

155 By-pass connections should be provided around the coolers in order to enable the plant to discharge its gas direct into the lines in case of leakage in the coolers.

156 All station piping should be "flexible," so as to permit operation with various systems of connection, and should be arranged in such a manner that the plant can be tested with ease.

DISCUSSION

C. N. Cross. There is one locality in the California oil fields which promises to become commercially important in the production of natural gas, viz., the Sunset-Midway district. A single

well has for the past two years been producing sufficient gas to supply the entire city of Los Angeles.

That this statement is not an exaggeration will be more readily understood when it is known that the initial rock pressure of the gas in the field is often more than 2000 lb. per sq. in. When these gas pockets are first opened the string of tools is often thrown from the well and over the top of a 90-ft. derrick. Sometimes large quantities of sand and boulders weighing 15 to 20 lb. each are ejected from the mouth of the well. Under such conditions there are very lively times getting the well under control. For lack of transportation facilities this immense quantity of gas is largely being lost.

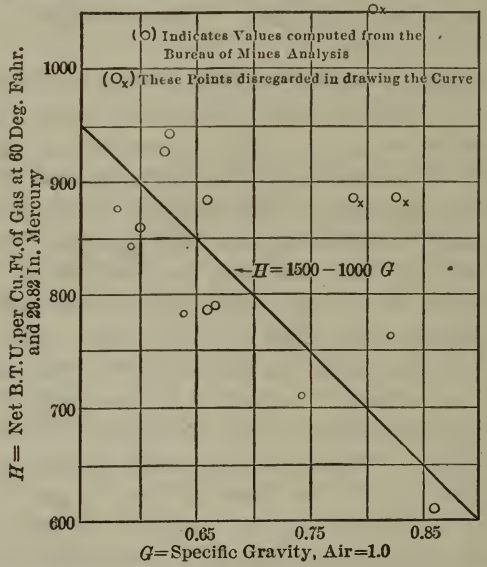

FIG. 14 RELATION OF CALORIFIC VALUE AND SPECIFIC GRAVITY FOR CALIFORNIA GASES

Noting the absence of California gases from Mr. Weymouth's table of analysis, the writer presents Table 4. The first five analyses were obtained in connection with the writer's work in the oil fields.

The sum total of the constituents of these five samples varies from 99.0 to 99.9 per cent. This irregularity is due to the fact that the measuring burette used could be read only to the nearest 0.2 per cent, and no attempt was made to determine the rarer gases. The percentage of nitrogen was read from the burette

TABLE 4 ANALYSES OF NATURAL GAS FROM THE CALIFORNIA OIL FIELDS. PER CENT BY VOLUME REDUCED TO 60 AND 29.82 IN. MERCURY

Location	Year	O_2	CO_2	(C_2H_4) (C_2H_6)	CH_4	N_2	Lower B.t.u. Junker Calorimeter	Lower B.t.u. by Analysis	Lower B.t.u. by Equation	Specific Gravity
Coalinga No. 1....	1911	....	19.20	....	79.0	1.8	667	709	754	0.7460
Coalinga No. 2....	1911	1.60	3.80	....	94.4	...	776	847	903	0.5974
Sunset No. 1.....	1911	....	22.80	10.7	65.9	...	670	762	678	0.8217
Sunset No. 2.....	1911	0.70	7.70	....	87.5	3.6	...	785	857	0.6434
Sunset No. 3.....	1911	1.30	0.60	3.10	92.3	1.7	921	877	918	0.5824
Analyses made by Bureau of Mines and Published in Bureau of Mines Bulletin, No. 19, 1911										
Santa Maria.....	1909	0.2	15.5	20.2*	62.7	1.4	...	884	675	0.8254
Santa Maria.....	1909	0.2	14.1	19.0	64.6	2.1	...	882	709	0.7910
Torry.....	1910	...	6.8	35.6	54.2	3.4	...	1053	695	0.8051
Coalinga.....	1910	...	11.1	....	88.0	0.9	...	789	836	0.6640
McKittrick.....	1909	...	30.4	1.0	66.2	2.4	...	610	638	0.8616
West Los Angeles...	1909	0.1	1.0	2.7	91.0	5.2	...	859	902	0.5978
Sunset.....	1909	...	10.5	....	87.7	1.8	...	787	838	0.6619
Fullerton.....	1909	...	1.7	9.5	86.7	2.1	...	929	876	0.6241
Fullerton.....	1909	...	1.4	10.9	85.8	1.9	...	944	873	0.6271
Kern River.....	1909	...	6.5	8.0	84.3	1.2	...	884	841	0.6594

* The remainder of this column C_2H_6 only.

There was a trace of H_2 in Sunset No. 2, and of H_2S in Coalinga No. 1 and Sunset No. 1.

instead of subtracting the sum of the other gases from 100 per cent. The remaining ten samples were analyzed by the Bureau of Mines and published in the Bureau of Mines Bulletin, No. 19, 1911. In every case the total of these analyses is 100 per cent.

The peculiar feature of the California gases is the unaccountably high percentages of carbon dioxide present. The fact that California oils are of an asphaltic base may in some way be responsible for the large percentage of carbon dioxide. Roughly, the straight line law connecting the calorific value and the specific gravity, shown by Mr. Weymouth, holds for these gases, but the slope of the line, Fig. 14, is reversed due to the presence of the carbon dioxide in large proportions.

The Author. The analyses and discussion of California gases given by Mr. Cross constitute a valuable addition to that portion of my paper dealing with the properties of natural gas.

Equations [4] and [5] in the original paper were constructed on the assumption that the chief influence affecting the specific gravity is the variation in the relative proportions of methane and ethane in the gas, and increase in the relative ethane content being accompanied by an increase in both specific gravity and heating value. When, however, the proportion of inert gases reaches the high values found in the California analyses, they exert a predominating influence on the specific gravity and thus reverse the slope of the line showing the relation of heat content to specific gravity, as Mr. Cross points out.

No. 1350

NEW PROCESSES FOR CHILLING AND HARDENING CAST IRON

By Thos. D. West, Cleveland, Ohio

Member of the Society

Conditions are occasionally such that an "inside chill" is produced in castings, by which is meant that a casting with a gray or soft exterior may have the interior, or portions of it, composed of a hard, white or chilled iron, as shown in Figs. 2, 3, 9 and 10.

2 While inside chilling is claimed to be produced by hydraulic pressure, this is not the inside chill which has puzzled foundrymen, and prior to the mechanical creation of inside chill by the author, no one, as far as he knows, has explained how it could be produced at will. The discovery of how this can be done is due largely to experiments which the author has been conducting during the past two years with a view to overcoming the defects now existing in chilled car wheels, and this investigation led to other lines of research, as will be seen herein.

3 Most of the experiments were made at The West Steel Casting Company, Cleveland, Ohio. This company makes castings by the converter and crucible methods, the former requiring a cupola similar to that used in iron founding, and so permitting the casting of chillable metals. The irons used were approximately of the following composition: Carbon 2.75 to 3.25; silicon 1.75 to 2.00; sulphur around 0.06; manganese and phosphorus about 0.04. In cases where a more chillable metal was desired, small portions of stick or powdered sulphur were dropped on top of the molten metal when in the hand ladle.

4 In Fig. 1 is illustrated the experiment that created mechanically an inside chill. This gives views of the mold used in casting test specimens in open sand of the size seen in Fig. 5. The bar *A* was cast

Presented at the Spring Meeting, Cleveland, 1912, of The American Society of Mechanical Engineers.

against a chiller block as at E, while bars B, C and D were surrounded with rammed sand. The bars having been poured, the gate connections F, G, H and I were broken and the sand around C and D removed as soon as their solidification would permit. When it was thought C and D would stand the pressure of tongs, they were lifted quickly and C was doused into a pail of water, while D was broken by an assistant to display conditions of its interior. The immersed bar C having cooled to a dark color, it was taken from the water and broken, and displayed an inside chill, such as is seen at E',

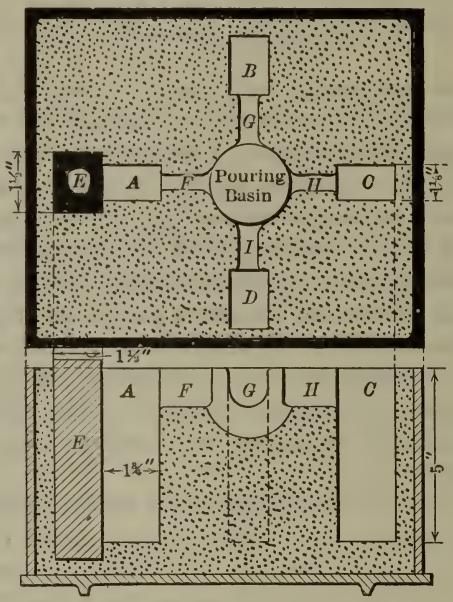

FIG. 1 MOLD FOR TEST BARS THAT SOLVED INSIDE CHILL PROBLEM

Fig. 9. The bar D showed an interior condition bordering on semi-molten metal. The chilled bar A showed that the chiller E had chilled the surface of A about $\frac{1}{2}$ in. deep, while the bar B had a nice gray fracture. Bars A and B were not removed from their molds until nearly cold.

5 The philosophy of mechanically creating an inside chill exists in the outside body of a casting cooling slowly enough to allow the carbon to take a graphitic form, while the inner body, which is not holly solidified, is cooled so rapidly that the carbon is held in the combined form, similar to the way it is held in the molten metal.

The graphitic state of the carbon in the solidified body causes the iron to be gray and soft, while the combined state causes it to be hard and white, or chilled.

6 The ability to create an inside chill by strictly physical manipulation led to the belief that it might be practicable to increase the depth of a chill beyond what present chillers can do, with like irons. Especially was this thought practicable for such castings as chilled car wheels and rolls. Several methods were devised for testing purposes, but finally those in Figs. 4, 7, 8, 13 and 14 were adopted for the chief researches. These devices admitted of a wide range for experiments and as a rule led to satisfactory results.

VARIABLE CONDITIONS AFFECTING CHILLING

7 As many who are interested in founding are not cognizant of what is involved in the chilling of cast iron, the following recital is given of actions that take place and of conditions that exist in the process of chilling:

a Iron is chilled prior to any formation of graphite.. A chill may be made harder by continuing to cool it while the adjoining metal is still in a semi-molten state or very hot; as by this action any backward annealing to soften the chill is more or less retarded.

b Under like conditions, the lower the silicon and the higher the sulphur and carbon, the deeper the chill.

c Under like conditions, the chill will be deeper the smaller the area of the cross-section to be chilled.

d Under like conditions the less the thickness in the chiller below what can be utilized, the less the chill in the casting.

e The longer the casting remains in close contact with its chiller while its metal is in a chillable state, the deeper the chill.

f The more fluid or the hotter the metal used in pouring a chiller mold, all other conditions being the same, the deeper the chill, from $\frac{1}{8}$ in. to possibly $\frac{3}{8}$ in.; and the greater the chillable nature of the metal, the more pronounced this effect.

g Not all grades of cast iron are chillable. It requires, as a rule, iron of a general carbon, having less than 2 per cent of silicon and above 0.06 per cent of sulphur.

h On account of the variable percentages of silicon, sulphur and carbon required to produce chilled castings, ranging from 1.75 down to 0.50 for silicon, from 0.10 down to 0.05 for sulphur, and from 4.00 down to 2.50 for carbon, it can be seen that there must be what are commonly called "grades" in chillable irons.

MOLD FOR MAKING COMPARATIVE CHILLING TESTS

8 In order to make comparative tests, a twin-chiller mold was

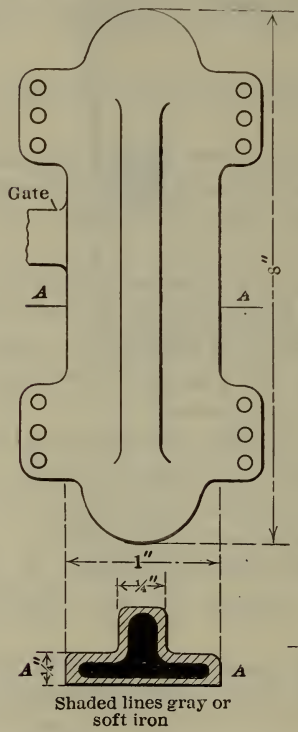

FIG. 2

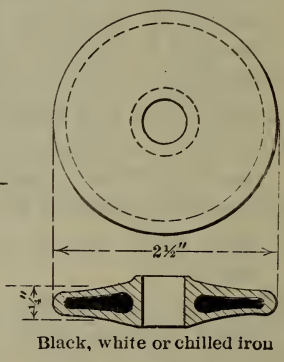

FIG. 3

CASTINGS WITH GRAY EXTERIOR AND CHILLED INTERIOR

designed, shown in Figs. 4, 7 and 8. By having the two molds combined so that they could be poured from the same basin, it was possible to make all the conditions alike in both, except the one which it was desired should vary for the particular test under way. The mold was further designed to produce conditions similar to those

existing when casting chilled car wheels and rolls. In making these latter castings the contraction of the chilled crust and the expansion of the chiller create a space between the exterior body of the casting and the interior face of the chiller. The ability to create a space as above in this experimental mold and to apply a heat-absorbing element to it in connection with other actions or treatment referred to later on, made it possible to conduct a large number of very satisfactory comparative tests.

9 In order to obtain a variable action so far as the chillers *P*, Fig. 4, were concerned, the braces *M* were removed and the wedges *N* driven down. If it were desired to have the space *K* in both molds, two persons were employed to work in unison at their respective ends. The wedges *N* would not be knocked down until the metal filling the mold showed evidence at the top edge of the bars of having solidified sufficiently to produce a self-sustaining crust. It was rather remarkable in the air-cooling tests, to be described later on, how clearly the rate of cooling and inner solidification could be judged by the changing color of the hot metal at the top edge of the bars next the chillers.

10 In casting bars intended to be kept in close contact with their chillers, either the runners connecting the pouring basin with the mold must be broken as soon as the metal in them has solidified sufficiently to permit such action; or care must be taken when pouring the molds not to fill them any higher than the level of the bottom of the pouring runners. By these means the disturbing influence due to the contraction of the runners and pouring basin is avoided.

11 Fig. 5 shows the form and size of the pattern used for molding the bars, while Fig. 6 is that of the manipulative chiller *P* employed in these tests. Each of these chillers weighs 3 lb. 14 oz. and the bars made from the pattern, Fig. 5, averaged 3 lb. and 4 oz. As Figs. 7 and 8 show devices for other purposes than those displayed in Fig. 4, comments on them are deferred.

FIRST EXPERIMENTS FOR DISCOVERING VALUE OF HARDENING MATERIALS

12 In starting to use the twin-chiller molds shown in Figs. 4, 7, and 8, the first manipulation was to draw back the chiller *P* of one mold about $\frac{1}{4}$ in. from the face of its bar casting as soon as the latter had solidified sufficiently to have a self-supporting crust facing the chiller, while the companion bar was left in close contact with its chiller until of a dark color (see Fig. 4).

13 In various tests a $\frac{1}{4}$ in. space K, Fig. 4, was formed by moving back the chiller, and the effect tried of quickly packing different materials into this space upon the backward movement of the chiller. The materials tried at different times were fine sand, coke dust, charcoal, powdered manganese, bone dust, hardening powder and poisonous cyanide. These tests were made with a view to determining whether such filling of the space would increase the depth of chilling in a bar, or cause it to be any harder than when the chiller remained in close contact with the bar until the latter was nearly cold.

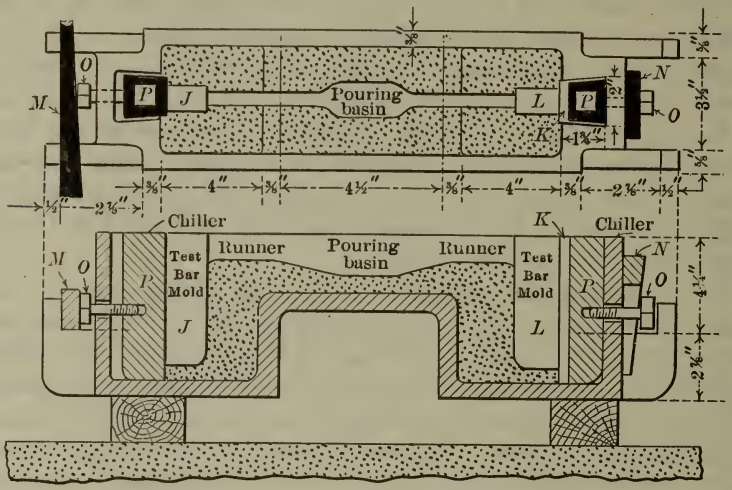

Fig. 4 Mold for Making Chilled Twin Specimens for Fracture Tests

14 No noticeable effects were produced by the use of these intended hardeners. The thickness of chill of the bars kept in close contact with their chillers exceeded that of the treated bars by from $\frac{1}{4}$ to $\frac{3}{8}$ in. This was to be expected, as the withdrawal of the chillers P to create a packing space broke the contact of the chillers with their bars at the crucial moment, so far as their effectiveness was concerned in increasing the depth of the chill.

SECOND EXPERIMENT FOR DISCOVERING VALUE OF HARDENING MATERIALS

15 In this series of tests, both chillers were pulled back at the same moment to create spaces adjacent to both bars, as at K, Fig. 4. The tests were to discover if any greater depth of chill or hardening

was produced by packing one of the spaces quickly with the materials used in the first experiments while the other space was left open. The series showed the chill to be from $\frac{1}{16}$ in. to $\frac{1}{8}$ in. deeper in several of the treated bars than in the non-treated bars.

16 Tests for hardness, except for the cyanide, showed a gain of 6 to 10 per cent by the scleroscope. One cyanide test with companion all-chilled bars averaged 68 for the treated and 70 for the non-treated; while with another set of companion bars which were not chilled the non-treated bar gave 58 and the treated bar 50 on an average. This softening action of the cyanide may have been due to the fact that the bars were not cooled in water after treatment, as is necessary when case-hardening steel with cyanide. A

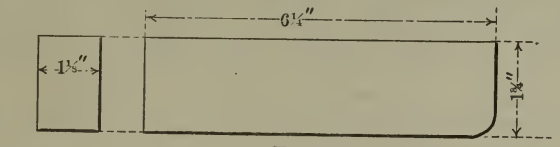

FIG. 5

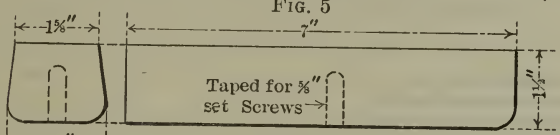

FIG. 6

FIG. 5 FRACTURE TEST BAR PATTERN; FIG. 6 MANIPULATIVE CHILLER

peculiar effect of the cyanide is to produce a denser and a finer crystallization of the treated chilled face about $\frac{1}{8}$ in. thick.

IMMERSION EXPERIMENTS

17 For this series, open sand molds of the plan shown in Fig. 1 were used, and by the aid of an assistant, two of the four bars were removed as soon as the crust solidification permitted. One of these bars was immersed and moved around in a pan of mud, which in some tests had salt mixed with it, while the second bar was simultaneously submerged in a pail of water. These tests were duplicated by having the pan nearly filled with a high fire-test oil in place of the mud.

18 Tests made by the second use of the open sand mold, Fig. 1, showed that the mud or oil cooling had little or no effect in chilling, on the average, while the water cooling was radical in its action. A', B', C', and D', Fig. 9, give a fair illustration of these tests. A' is the chiller-cooled bar; B' the all-sand one; C' the mud-cooled bar

and D' the one placed in water. Bars A' and D' were both left in their molds to cool naturally.

19 The bar P' displayed an inside chill as seen by the white appearance at D' and the gray corners and surface indicated by M'. The best sample of this effect, which was closely duplicated in several tests, is seen at E' in connection with the chiller-cooled companion bar F', which latter was cast against the face of the chiller E, Fig. 1, the same as bar A' in Fig. 9. The bar F' showed that a harder grade of iron was used than for A'.

20 The writer's experience in producing this inside chill mechanically, demonstrates it to be a sensitive process partaking of numerous forms, yet all verifying its practicability.

21 It is not improbable that castings, or sections of them, will eventually be produced in a regular manner for commercial use, having a gray or mottled exterior crust, while the adjoining or interior body of metal will be of a mottled, chilled or white iron. The writer can conceive of castings in which this combination of hardness might prove useful, but as his ideas might be considered visionary, he refrains from mentioning them.

TABLE 1 SCLEROSCOPE TESTS FOR HARDNESS *

Set Number of Tests	Set 1	Set 2	Set 3
Average of air-cooled chilled bars	56	65	61
Average of natural-cooled chilled bars	48	58	53
Average of increase in hardness effected	8	7	8

*The writer is under obligations to the kindness of Mr. Walter D. Sayle, president of the Cleveland Punch & Shear Works, Cleveland, O., for the scleroscope tests given herein.

HARDENING A CHILLED BODY WHEN HOT BY IMPINGEMENT OF AIR AGAINST ITS SURFACE

22 Special attention was paid in this series of tests to the hardening effect of air applied directly to the hot chilled face of a bar. For tests of this character it is essential that the depth of chill in the comparative bars be of the same thickness. This condition was obtained by admitting the air to the treated bar only after it was thought the inner metal had all solidified, so that it could not be held responsible for any variation of depth of chill in the treated bars. Only three specimens of this series were tested by the scleroscope, as given in Table 1; but from filing and grinding tests made as a check on the results it is evident that a chill's hardness can be very materially increased by applying air, etc., as herein described.

23 Tests of the gray sides of the chilled bars for the second and third sets in Table 1 gave 52 and 42 respectively, showing them to average 25.4 softer than the chilled sides which had received the air treatment.

EFFECT OF AIR SATURATED WITH WATER

24 This series was conducted with a view to learning whether air saturated with water might be more effective than air alone in creating a chill, or in hardening. There was some difficulty at the start in obtaining satisfactory saturation, but this was finally secured by the use of a device which made it possible to vary the proportion of air or water as desired.

25 The series was instructive in demonstrating that so far as the chilling was concerned there was little to be gained by the saturated air, or what was gained would be secured by the use of a greater volume of air alone, so that the water could be dispensed with. This is not, however, to belittle the effectiveness of saturated air as a hardening medium, for among the tests in which saturated air was strongly applied, one gave 70 for the treated bar, and 52 for the non-treated bar, a gain of 34.6 per cent due to hardening.

SOFTENING EFFECT OF ANNEALING CHILLED IRON

26 Specimens of this series of tests were cast in the twin-chiller mold, as arranged in Fig. 8, without applying any air cooling to either chillers. This gave the same depth of chill in both bars. Several sets were cast and annealed, but only one set was tested by the scleroscope. Other sets showed by filing and grinding that they checked closely with those tested by the scleroscope. The annealing was done as follows: The bars were taken out of their mold together and one left in the open air, of about 70 deg. fahr., while the other was laid on the bottom of a hot crucible furnace, the oil fire having been shut off, and left there about twelve hours.

27 The chilled side of the unannealed bar tested at 68 and the gray side at 45. The chilled side of the annealed bar tested at 47 and the gray side 35. This gave a difference for the chilled sides of 21 and for the gray sides of 10.

RELATIVE EFFICIENCY OF WARM AND COLD CHILLERS

28 In order to be assured of correctness in his researches, the author undertook to determine the efficiency of different metals, and of different thicknesses of metal, for chillers and the relative efficiency of warm and cold chillers.

29 To determine the latter, two chillers, both alike, were used, one of which was heated to various temperatures, from one bearable to the hand to one that would burn the flesh, while the other chiller was kept at the temperature of the atmosphere. The first three sets of these tests were made on a day when the thermometer registered 62 deg., and the second set of four tests when it was 55 deg.

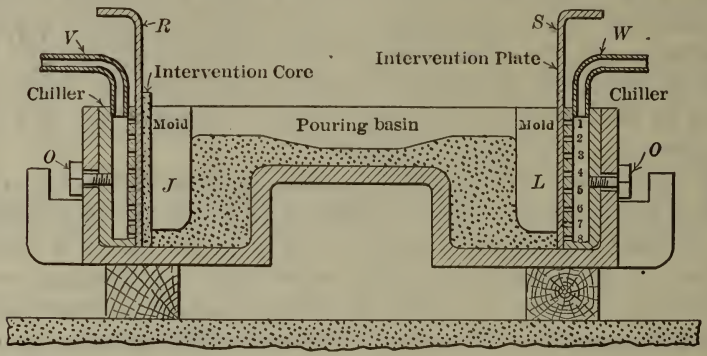

FIG. 7 INTERVENTION PLATE SYSTEM FOR AIR COOLING AND CHILLING OF MOLTEN METAL

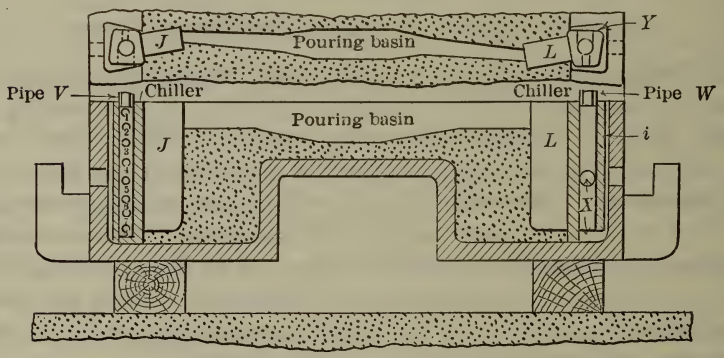

FIG. 8 CHILLER MOLD FOR TESTING CHILLING EFFECT ON CASTINGS BY COOLING CHILLER

The tests showed from $\frac{1}{16}$ in. to $\frac{3}{16}$ in. greater depth of chill in the bars having the cold chiller than those having the heated ones. Having recently read the claim that a warm chiller would chill deeper than a cold one, it was thought necessary to learn whether such was a fact, as it is unreasonable to expect such a result.

EFFICIENCY OF DIFFERENT THICKNESSES FOR CHILLERS

30 To determine the effect of different thicknesses for chillers,

tests were made by the writer at the Cleveland branch of the National Car Wheel Company, as well as at the foundry of The West Steel Casting Company. For this series open sand molds were used, as seen in Fig. 11. The chillers as shown were respectively $\frac{1}{4}$ in., $\frac{1}{2}$ in., $\frac{3}{4}$ in., 1 in., 2 in. and 3 in. thick; and all 2 in. wide by 9 in. deep.

31 The test specimens cast against these chillers in the wheel foundry were $1\frac{7}{8}$ in. by $2\frac{1}{2}$ in. and 8 in. long, and at the steel foundry, because of having a less chilling metal, they were of the size shown in Fig. 5. The top row of fracture views, Nos. 1 to 6, Fig. 10, is a fair representation of results obtained. The thickness of chill in the samples shown in Fig. 10 is approximately as marked. A study of these samples demonstrates that the efficiency of chillers in general use is far from being in accord with their thickness. The 3-in. chiller, for example, produced but $\frac{3}{16}$ in. more depth of chill than the 1-in. chiller.

32 Aside from their value in other ways, these tests suggest the advisability of makers of chilled rolls, car wheels, etc., trying steel metal chillers in place of the much heavier cast-iron ones. Steel chillers need be made only of the thickness required for efficiency in cooling and would still be strong enough to resist the contraction and expansion strains to which chillers are subjected. There is a liability of difficulty being encountered in using steel chillers on account of the steel warping much more than cast iron through repeated heating. By using ribs or giving a special form to steel chillers this objection may be greatly reduced, if not wholly overcome.

EFFICIENCY OF DIFFERENT METALS FOR CHILLERS

33 It was important, at least to the writer, to know whether any difference existed in the efficiency of chillers of gray, white or all-chilled cast iron or of steel or wrought iron. Tests were made with these different metals at both foundries. Two sets of chillers were used, one 2 in. square and the other 1 in. by 2 in., all close to 9 in. long. Aside from grinding one face of these chillers to remove the scale and make them smooth, as with all other chillers used in the experiments of this paper, they were chipped or ground at their lower ends if needed, to make them all of the same weight. The plan of the open sand mold used is shown in Fig. 12. The test bar patterns were of the same size as used for the tests of Fig. 11, about four sets being cast from each size of pattern. While slight differences in the thicknesses of the chills resulting from chillers of the different materials seemed to indicate that one or another of the

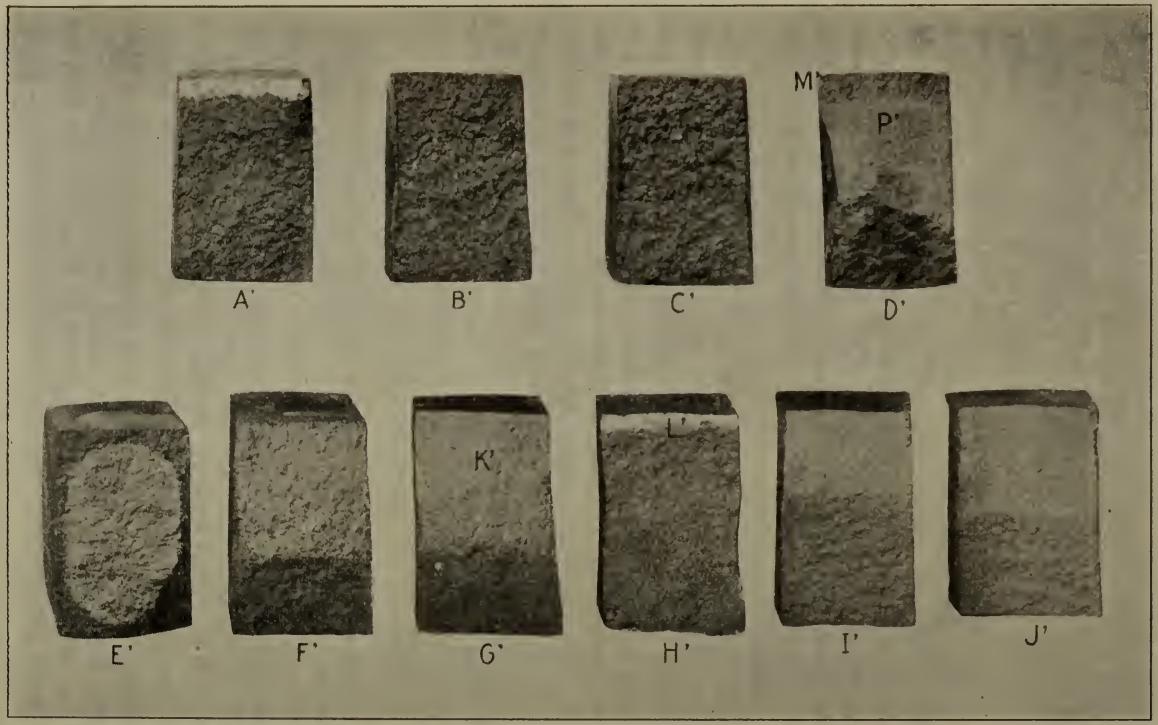

FIG. 9 SAMPLES OF TEST BARS. UPPER ROW SHOWS RESULTS OF IMMERSION EXPERIMENTS. IN THE LOWER ROW *E'* SHOWS AN INSIDE CHILL, *F'* A CHILLER-COOLED BAR AND THE REST ILLUSTRATE THE SUPERIORITY OF AIR CHILLING OVER METAL CHILLERS

FIG. 10 SAMPLES OF TEST BARS. UPPER ROW ILLUSTRATES EFFICIENCY OF CHILLERS OF DIFFERENT THICKNESSES; LOWER ROW, RESULTS WITH SAND-FACED MOLD AND AIR-COOLED BARS CONTAINING VANADIUM AND TITANIUM

materials produced the deepest chill, there was so little practical difference in the results, taking an average of all the tests, that none of the chillers could be rated as being decidedly better than the others.

VALUE OF COOLING CHILLERS

34 The chief information sought in this series of tests was whether, when a casting contracts away from a chiller, the depth of chill is increased by cooling the chiller, and at what point the cooling

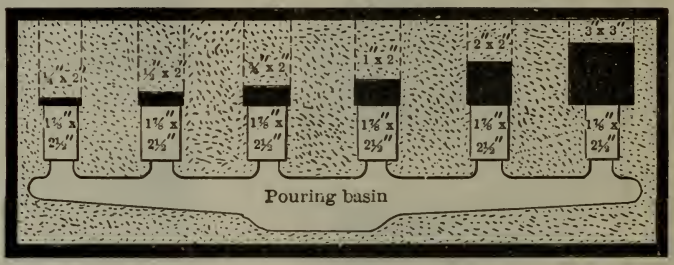

FIG. 11 TESTING EFFICIENCY OF DIFFERENT THICKNESSES OF CHILLERS

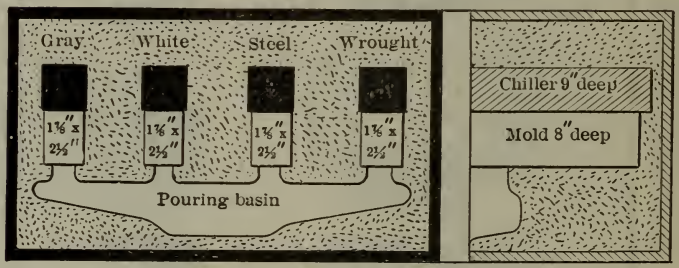

FIG. 12 TESTING EFFICIENCY OF DIFFERENT METALS FOR CHILLERS

ceases to have this effect. The chillers for these tests were arranged so as to have the orifices 1 to 8, Fig. 7, turned away from the face of the mold as in Fig. 8. This gave a solid surface to the body of the chiller fronting the mold, and prevented the air used to cool the chiller from impinging on the hot face of the bar. For the first tests, intervention plates *S* were used for fronting each chiller, after the ideas displayed in Fig. 7. As soon as solidification of the metal at the face of these plates would permit, they were pulled out simultaneously to create a space between the bars and their chillers, as seen by the opening at *K*, Fig. 4. The plates *S* removed, air of about 50 lb. pressure was admitted to one of the pipes, as at *W*,

Fig. 7, which passed down the bore of its chiller to find exit at *X*, and at the orifices 1 to 8 to the exterior of the chiller; from whence it passed upward and escaped to the external atmosphere from around the opening *Y* in the plan view of Fig. 8.

35 In the use of chillers for car wheels, rolls, etc., the contraction of the casting and expansion of the chiller leave a space between the two, and these tests were therefore well adapted to demonstrate whether internally or externally cooled chillers used for this class of castings would produce a deeper chill than those not cooled. Half a dozen tests were made without finding any practical difference in the depth or character of the chill produced by the air-cooled chiller bars and that of their companions which were not cooled.

36 Following these tests, fully six more were made with the plates *S* omitted on both sides, so as to have the bars cast directly against the face of their chillers, as displayed in Fig. 8. Here both bars remained in close contact with their chillers until cooled to a dark color. In casting these bars air of about 50 lb. pressure was admitted to one side only, passing down the pipe *W* and escaping at the lower exits seen at *X*, and thence passing upward to the external atmosphere through the space i. This method proved practically no more effective in increasing the depth of the chill than that just described and appears to confirm the tests illustrated in Fig. 11, demonstrating that there is a limit to which the thickness of a chiller affects its efficiency. They forcibly illustrate the fact that it would be unreasonable to expect any benefit from a heat-absorbing medium passing rapidly through the internal body of a chiller or over its outer external surface when a space existed between the casting and the chiller.

CHILLING PRODUCED BY SAND-FACED MOLDS

37 It was desired to ascertain whether a chill could be created by air under pressure when prevented by a sand coating from getting directly at the hot surface of a casting. To test this the methods illustrated in Figs. 7, 13 and 14 were used, the last two of which were experimented with at both foundries.

38 In using the mold shown in Fig. 7 an intervention core was employed, as seen on the left side. This core was about $\frac{1}{4}$ in. thick and well wired so that the head pressure of the molten metal could not break it, when the plates *R* and *S* were pulled out together

after pouring the molds. The first test of this series demonstrated the porosity of a sand mold's surface. Although this was a very hard core, the 50-lb. air pressure used carried the air through it and would have blown all the metal out of its mold had not the valve been closed. The companion bar withstood the air pressure for the reason that the plate *S* had formed a chilled crust on the face of the metal in the mold before its removal. Further tests with these cores under different air pressures showed that the cores prevented

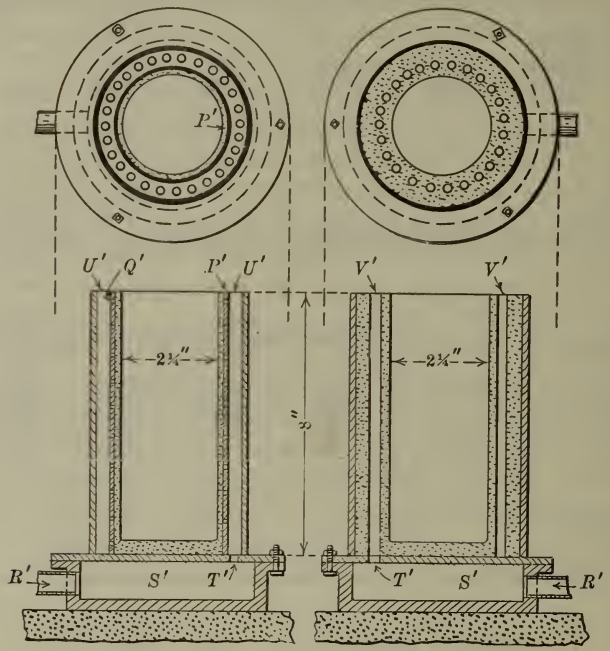

FIG. 13 PIPE SAND-COATED AIR-CHILLING MOLD

FIG. 14 ALL SAND AIR COOLED MOLD

any chilling action, while on the other hand the air was very effective on the opposite side.

39 The difficulty with this core method lies in the fact that air under sufficient pressure to penetrate the core and carry off heat in time to create an internal chill will pass clear through the molten metal instead of forming a crust and chilling it. This led to the de-

signing of the method seen in Figs. 13 and 14. By a study of the sand-coated pipe P', it can be seen that any pressure up to 200 lb. and over can be employed without causing the air to impinge against the molten metal, while at the same time it can be more effective than the core itself in conveying heat to its outer surface and then to the atmosphere. The sand coating used on this pipe was but $\frac{1}{8}$ in. thick and thoroughly dried. To carry off the gases from the sand the iron pipe was closely perforated with $\frac{3}{32}$-in. holes as at Q'.

40 Air under 60 lb. pressure entered at R' to the chamber S' and passed up through the holes T' surrounding the pipe P'. The air in the chamber U' was free to absorb heat from the pipe P' and to carry it rapidly to the atmosphere. The molds were poured without any cope or covering, care being taken to fill them only within $\frac{1}{2}$ in. or so from their tops. In about 10 seconds after pouring the mold, the air was admitted and kept in action until all the metal was thought to have solidified. A fair illustration of results is seen at M', Fig. 10. An all sand molded, non-treated companion bar was always cast from the same ladle that poured the treated bar. A sample of this is seen at N' on the right of M'.

41 There is little doubt but that the pipe P' acted as a chilling agent without the use of air, as there was only $\frac{1}{8}$ in. thickness of sand between it and the face of the casting. In fact, a test made without the air showed that the pipe aided the chilling, since it gave a density to the crust of the casting. Again it is to be kept in mind that as the diameter of the casting was only $2\frac{1}{4}$ in., its contraction would not be sufficient to create a visible space between its outer body and the face of the mold, as is generally created in casting chilled rolls, car wheels, etc.

42 In the all sand mold, Fig. 14, which was also dried, the air passed up through holes T', which were $\frac{1}{4}$ in. in diameter and about the same distance apart, all around the circumference, as seen by the plan view. The air escaped freely around the top at V'. The holes T' had but about $\frac{3}{16}$ in. thickness of sand between their inner exterior and the face of the mold. Castings produced by this method showed a dense exterior or crust of from $\frac{1}{8}$ in. to $\frac{3}{8}$ in. thick, and in some instances were slightly mottled. Only in one case was there any display of chill, and this was of an irregular character $\frac{1}{8}$ in. to $\frac{1}{4}$ in. thick, created inside of a gray crust about $\frac{1}{8}$ in. thick. In reality this was an example of inside chill, one of the factors sought

in these experiments being to learn if by such methods it could be produced at will and if it was controllable.

43 It was intended also to form holes as at V' with very thin pipes drilled closely full of $\frac{3}{16}$-in. holes, and if a steady pressure of 100 to 150 lb. of air could have been secured, further efforts to learn the practicability of obtaining an inside chill with such, or similar, methods would have been tried.

44 Study of the principle embodied in Figs. 7, 13 and 14 will suggest ideas and ways and means by which grades of soft iron, aside from chilling iron, can be made more dense or crust-hardened; also by which castings or sections of castings may be cooled to prevent contraction cracks and shrink holes.

EFFECT OF A JOINT, OR SPACE, ON TRANSMISSION OF HEAT

45 It is evident that in constructing chillers to be cooled by air, etc., sufficient consideration has not been given to the question of

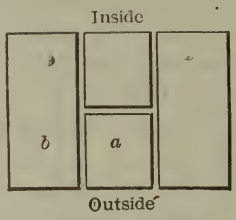

FIG. 15 ILLUSTRATING RETARDATION OF FLOW OF HEAT CAUSED BY JOINTS

transmission of heat. An excellent article by Carl Hering on The Flow of Heat through Furnace Walls [1] might be read with advantage in connection with this paper. Its author offers among other features the illustration seen in Fig. 15 with the following comments: "Moreover, the writer noticed recently in an electric furnace that the temperature of the brick at *a* in the sketch was considerably cooler than at *b*, the difference was so great that it was easily noticeable to the touch. This was no doubt due to the joint which separated the one from the inside of the furnace."

46 The principle illustrated in Fig. 15 is similar to that involved where such spaces exist as at *K*, Fig. 4, which greatly retard the absorption by the chiller of heat from the castings. It all emphasizes the utility of filling such a space as *K* with a heat-absorbing medium that can be moved swiftly from an inlet to an outlet to carry off heat for cooling, hardening or chilling purposes.

[1] Metallurgical and Chemical Engineering, September 1911.

47 It is believed that a study of the conditions will show that a rapid passage of air or other heat-absorbing media through a space as at *K*, Fig. 4, or on the surface of a casting, is very efficient for the purpose of extracting and conveying heat quickly both from the face and adjoining body of a hot casting.

TESTS ON HEAT CONDUCTIVITY OF SAND, IRON AND AIR

48 These tests were made for the purpose of ascertaining the heat conductivity of molds, composed of a sand body, and again of iron, as in chillers, instead of having a heat-absorbing medium impinge directly against the hot surface of a casting with exits for the rapid escape of the medium to the atmosphere.

49 The tests for sand conductivity were made by constructing a dry sand core $1\frac{1}{2}$ in. sq. by $7\frac{1}{2}$ in. long, with a $\frac{5}{8}$-in. port hole lengthwise through the center. This core had part of the outer face of one end cut away to provide an exit for the air forced through the interior of the core. When placed in position the core appeared practically like the chiller seen on the right of Fig. 8.

50 The tests for iron conductivity were made by having a chiller with solid face and other conditions of its position as seen on the right of Fig. 8.

51 In conducting these tests for the sand and iron, as well as with air impinging against the surface of the hot castings, only one end of the flask for twin molds was used. The mold *L* was formed with the same pattern, Fig. 5, as used for making all other bars. The molds were poured by a direct flow of the metal from the lip of the ladle. The top of the bars having been covered with sand and a plate to confine their heat so that the thermometer used would read correctly, the air-flow was started down the interior of the core or chiller and found an escape at the opening *X* and up the sides *I* to the top around *Y*, thence to the atmosphere, as in Fig. 8.

52 To obtain the temperature of the escaping air, the bulb of a thermometer capable of registering 500 deg. was held directly above and resting on the surrounding sides of the open space at *Y*. An assistant recorded the time and the varying temperatures of the escaping air at the first 15 and 30 seconds and afterwards each minute, as seen by lines 5 and 6 of Table 2. At the end of 10 minutes the bulb of the thermometer was held at one end of the inlet pipe after it was disconnected in order to obtain the record of line 4. At the end of 2 minutes, and after the plate and sand had been removed from the top of the bar, the bulb of the thermometer was placed so

that its frame end rested on the middle of the top end of the bar for 3 minutes, that the natural radiation of heat from the bar, seen in the last line of Table 2, might be recorded.

53 The variations in pressures and temperatures seen in lines 3 and 4 are due to changes in the speed of the compressors and to the amount of air being taken from the tank for other purposes. The seeming inconsistency of the temperature dropping at the start to only 80 deg. and 94 deg., while that which came from the tank of the compressors is 134 deg. and 124 deg., as in line 4, is due to the temperature of the core and chiller at the start being that of the atmosphere, as seen in line 2. For these reasons, the ingoing air is for a few moments reduced in temperature.

TABLE 2 TEMPERATURE TESTS MADE OF THREE CASTS, OCTOBER 23, 1911

		Sand Core	Chiller	Face of Bar
1	Body cooled with air			
2	Temperature of atmosphere, deg. fahr.	52	52	52
3	Pressure of air used, lb.	55	50	50
4	Temperature of tank air, deg. fahr.	134	124	130
5	Temperature escaping air at 15 sec.	80	94	...
6	Temperature escaping air at 30 sec. deg. fahr.	100	112	200
7	Temperature escaping air at 60 sec. deg. fahr.	116	142	290
8	Temperature escaping air at 2 min. deg. fahr.	120	162	310
9	Temperature escaping air at 3 min. deg. fahr.	126	182	290
10	Temperature escaping air at 4 min. deg. fahr.	130	182	270
11	Temperature escaping air at 5 min. deg. fahr.	134	178	264
12	Temperature escaping air at 6 min. deg. fahr.	136	176	252
13	Temperature escaping air at 7 min. deg. fahr.	138	172	236
14	Temperature escaping air at 8 min. deg. fahr	140	170	222
15	Temperature escaping air at 9 min. deg. fahr.	140	166	200
16	Temperature escaping air at 10 min. deg. fahr.	138	162	192
17	Heat radiated from the bars 15 min. after they were poured, deg. fahr.	208	172	70

54 A study of Table 2 shows the sand to be the least effective as a conductor of heat while the iron is not very much better when compared to the conductive power of air applied directly to the surface of the hot bar, as recorded in the last column, line 17; from the moment the air impinged upon the surface of the hot bars its temperature rose and in less than 30 seconds after the mold was poured reached 200 deg.

FURTHER EXPERIMENTS UPON DIRECT APPLICATION OF HEAT-ABSORBING MEDIA

55 The following experiments were largely responsible for patents granted May 1912 and pending on direct-cooling and treatment processes by pressure or suction for chilling, hardening and

relieving internal strains in castings, etc. These are to be utilized wherever a space can be formed adjacent to a casting, either artificially or by the natural expansion of the chiller and contraction of the casting, also when the hot surface of a casting is freely exposed to the atmosphere or not surrounded by its chiller.

56 Two twin molds were used for the experiments, each having intervention plates placed as at *S*, Fig. 7. After the two molds were poured from the same pouring basin, and a crust was formed on the face of the bars, the two plates were quickly and simultaneously removed from the mold, forming spaces as at *K*, Fig. 4. For the first experiments air of 20 lb. to 30 lb. pressure was admitted to one of the pipes *W*, Fig. 7, and conveyed directly to the space created by the removal of the plates *S* through orifices 1 to 8 in the bore of the chiller. The fast darkening of the top edge of the face of the treated bar. compared with that of the companion bar which was cooling naturally, gave good reason to expect considerable difference in the depth of chill and in the density or hardness of the chilled face of the two bars.

57 The intervention plates *S* were 2 in. wide by $\frac{1}{8}$ in. thick. A slight coating of oil was given the faces of these plates next the bars to prevent their uniting with the metal, and to permit their being drawn out of the mold quickly at the right moment.

EXPERIMENTS WITH COOLED AIR

58 Experiments were further made with air cooled by passing through a pipe coil surrounded by a mixture of two parts of cracked ice and one part of salt. The temperature was reduced thereby from 85 deg. to 45 deg. but no greater chilling effect was discovered in the six tests conducted on this plan than in the first series with the air as it came from the tanks. This is as would be expected, since the reduction of the temperature of the air by 40 deg. is so small in amount compared with the temperature of 2000 deg. which it may be assumed the surface of the molten bar would have. An increase in the pressure or volume of the air would easily discount all that could be accomplished by lowering the temperature of the air to 40 deg. as noted.

SUPERIORITY OF AIR OVER METAL CHILLERS

59 Tests were also conducted with air at higher pressures. At 50 lb. pressure a chill was created for a depth of $1\frac{1}{8}$ in. in the air-cooled bar, whereas the naturally cooled bar had a depth of chill of only $\frac{1}{16}$ in., as seen respectively at *K'* and *L'* in the bars *G'* and

H', Fig. 9. The gray body of both of these bars displayed a fine texture bordering on a mottled state. This test removed every possible doubt of the efficiency of air-cooling for chilling. Half a dozen or more of these tests were made before taking up others, and they all verified the results of the first tests.

60 Later on tests of the same character were conducted by having titanium and vanadium in the metal. Two sets of these samples are shown at W' and X' and again at Y' and Z', Fig. 10. Here as in G' and H', Fig. 9, the air was by far the most effective in chilling. The air-treated bars showed about $\frac{7}{8}$ in. depth of chill whereas their companions had but about $\frac{1}{16}$ in. chill.

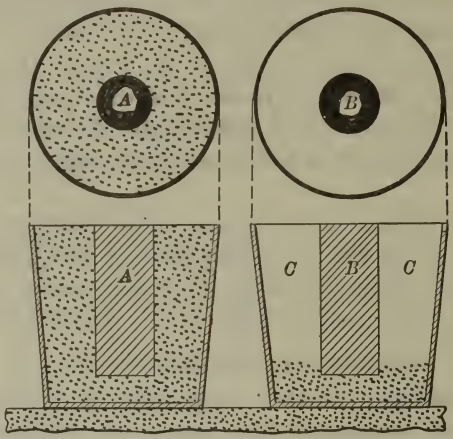

FIG. 16 SPECIMEN BAR CAST IN OPEN SAND AND AFTERWARDS FREED OF SAND AND SURROUNDED BY WATER

61 In line with these tests a series was made to determine whether air chilling was more effective than chilling by means of a solid chiller held in close contact with its bars. Samples of bars contrasting these two methods are seen at I' and J', Fig. 9, the air-cooled bar at J' having $\frac{3}{4}$ in. chill, while the other at I', produced by the close contact chiller bar, has $\frac{9}{16}$ in. chill.

62 Chilling of iron must be accomplished prior to the formation of graphite, and with like irons the quicker and more penetrating the cooling action, the deeper and harder the chill. The direct application of a heat-absorbing medium to the surface of a hot casting, as soon as contact with its chiller is broken, or a crust is formed, provides means at a critical moment which can not but be of material benefit in

increasing the utility of cooling, densifying, or chilling and hardening of chillable and other grades of cast iron. It also provides means for securing a softer, or lower chilling and using a stronger iron in car wheels, etc., obtaining at the same time the desired depth of chill.

PRACTICABILITY OF CONTINUING CHILLING AFTER THE METAL SOLIDIFIES

63 It has always been thought impracticable to produce a chilling action in the process of cooling or chilling an iron, after the metal has once solidified. Later experiments by the writer show that such is not the case. There is a period of 20 to 30 seconds after the formation of a crust, according to the grade of iron, during which chilling can be conducted. This can be demonstrated by casting as follows: At *A*, Fig. 16, is shown a casting poured in open sand, while at *B* it has been freed of its sand, this being done about 2½ minutes after the casting was poured.

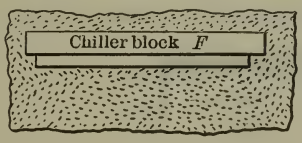

FIG. 17 MOLD FOR CASTING A CHILLED PLATE

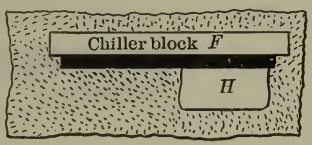

FIG. 18 MOLD WITH SPACE *H* FOR MOLTEN METAL

Space *C* was then immediately filled with cold water, kept running until the casting was cold.

64 Upon breaking the specimen it was found, if of a high chilling iron, to be a homogeneous body of all chilled or white iron with a discolored or reddish center. But if, instead of surrounding the specimen with water at the expiration of 2½ minutes, there were allowed to lapse 3 to 3½ minutes before doing so, the crust exhibited graphitic formation, while the interior body was found to exist in a mottled or all-white state, showing the inside chill to have been created.

TWO NEW PRINCIPLES IN CHILLING

65 These tests indicate the existence of two laws or principles that can be accepted as being positive in their action, as follows: First, cooling or chilling is effective in continuing or originally creating a chill in a casting for a period of 20 to 30 seconds after its molten metal has solidified. This permits a continuation of chilling, with castings like rolls and car wheels which

break contact with their chillers immediately after the formation of their chilled crust. By means of the writer's recently developed processes, castings of chillable iron can be given a deeper chill than if the power to chill a metal ceased at the moment of its solidification.

Second, graphitization having once taken place in the crust or body of a hot casting, no sudden cooling can restore the carbon to its original combined form, and only by remelting can it be so transformed as to have a chilled or white iron structure.

DIFFICULTIES ENCOUNTERED IN CREATING AN INTERNAL CHILL

66 With chillable irons any founder can produce a casting having an outside chill with a gray interior, but to produce one having a gray exterior and inside chill, or white body, is another proposition.

67 Mention has already been made of the sensitive nature of such a production. The variable conditions that must be considered and controlled to an exactness in order to create a perfect inside chill are as follows: (*a*) Temperature of the pouring metal;

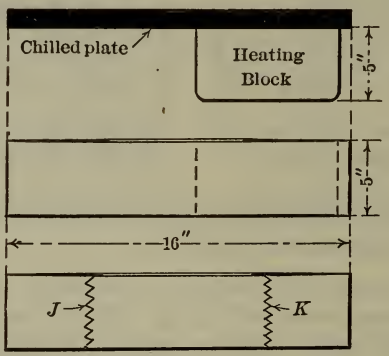

FIG. 19 CHILLED PLATE WITH HEATING BLOCK ATTACHED AND SEPARATED CASTING

(*b*) temper of the sand; (*c*) atmospheric conditions; (*d*) nature of the iron; (*e*) size of the specimen; (*f*) temperature of the water; (*g*) whether the specimen remains stationary in its mold to be cooled or is removed or moved about in a body of water.

68 When it is stated that, for an example, with the size of specimen seen in Fig. 1, there are only some 5 to 10 seconds during 1 minute when the perfect inside chill can be created, all students of this problem will realize that at present it is a hit-and-miss process. The writer has deemed it necessary to give all the above facts, so

that anyone undertaking to produce an inside or internal chill will not be led to affirm it an impractical achievement. It will be well to state that the writer is of the opinion, that when one can obtain a medium chilling iron in place of the extremes, such will be best for creating an internal chill.

INTERNAL CHILLING TESTS WITH HIGH SILICON AND SOFT CASTING MIXTURES

69 Tests made by the writer, assisted by Mr. W. J. Strangward, superintendent, at the Forest City Foundry & Manufacturing Company, Cleveland, Ohio, showed that the high silicon in their light work mixtures caused graphitization to take place almost immediately, if not at the moment of solidification, as the specimens exhibited no white or even mottled internal structure in any of the tests.

70 Other tests made at the Madison Foundry Company of Cleveland, upon supposed non-chilling mixtures with silicon around 2.0 and sulphur under 0.08 showed that these percentages of silicon and sulphur marked the division between chilling and non-chilling irons of the usual good working grade of foundry mixtures. The tests showed that if internal chilling could be produced with mixtures having from 2.0 to around 3.0 per cent silicon, there was a possibility that something was wrong with the physical or the chemical properties of the mixtures.

71 A feature of these tests with soft irons was that they showed a swelling in place of a shrinkage in their taps; also a greater fluidity of metal, or length of time before solidification took place. Although these bars were but about $1\frac{1}{2}$ in. in diameter they remained in a liquid state about as long as specimens $2\frac{1}{4}$ in. in diameter cast of the chillable car wheel metal.

STANDARDS FOR INTERNAL CHILLING TESTS OF HARD AND SOFT GRADES OF IRON

72 Founders and engineers interested in castings for machining, etc., might often utilize internal chilling tests to advantage as a means of determining whether there is anything in the chemical or physical properties of mixtures likely to cause chilled edges, hard spots, etc., instead of waiting for this to be found out in the machine shop.

73 The writer would suggest as a standard for such tests, bars $1\frac{1}{2}$ in. in diameter and 6 to 8 in. long for mixtures ranging from 1.25 to 3.5 per cent silicon and bars 2.4 in. in diameter of similar length for mixtures having from 0.5 to 1.25 per cent silicon.

74 In making internal chilling tests care must be exercised not to immerse a specimen in water until a self-supporting crust has been formed, or an explosion of liquid metal may occur. The use of a fair amount of intelligence and caution will guard against such dangers.

INTERNAL GRAPHITIZATION OF A CHILLED CRUST

75 Being desirous of knowing whether, after the greatest depth of a chill is created, it is possible for the intense heat of an internal body of semi-molten or solid metal to decrease the depth of a chill by graphitization, or "drawing a chill," the writer conducted the following experiments: An open sand mold having a chiller block *F* was used, as seen in Fig. 17. The open sand mold was formed by plate patterns 5/8 in. and 7/8 in. respectively, both being about 5 in. deep and 16 in. long. After the plates were poured and solidified, a space was dug out for about half their length, as seen at *H*, Fig. 18. This space was filled with molten metal, left in close contact with the chilled plates until cooled to a dark color. The molten metal was of regular car wheel mixture, and the tests were conducted at the National Car Wheel Company's plant, Cleveland, Ohio.

76 Tests were made with the plates at different temperatures from that at which the molten metal would fuse the face of the plate, down to temperatures at which the plates were of a dark color. Upon removing these plates and body blocks of metal from the molds, the chilled plates would be separated from the blocks of metal and broken at about the points *J* and *K*, Fig. 19, to display any contrast that might exist due to the treatment. Only in one case was the plate inseparable, and in this instance the plate and block were broken by a heavy drop block.

77 In all of these experiments a drawing of the chill in depth was displayed by reason of the hot molten metal causing a graphitization of the chilled face abutting it. The experiments made with the hottest plates showed the greatest effect, and such as to produce about a 25 per cent graphitization of the chilled plates face that abutted the hot metal. This effect was exhibited by a fairly uniform decrease of the graphitization down to the coldest plates, which showed but a slight effect of the treatment.

DISCUSSION

Henry M. Howe.[1] Mr. West found that, if after the outer part of a cast-iron casting has solidified and cooled at a normal

[1] Prof. of Metallurgy, Columbia University, New York.

rate to somewhat below its eutectic freezing point, thus becoming graphitic, the casting is suddenly withdrawn from the mold and quenched in water at a moment when the interior has already solidified, the inside will be of white iron. The fact that the inside of the casting is white, though the normal slow rate of cooling extended below the eutectic freezing point, shows that it solidified as white iron. Because the inside solidified as white iron, the outside evidently did also. The graphitization of the outside must have occurred in that phase of the cooling by which the gray outside had outrun the white interior at the moment when the quenching occurred.

Because the amount by which the cooling of the outside outran that of the inside must have been very slight, the writer infers that this graphitization occurred in a very narrow range of cooling, and presumably below the eutectic freezing point. The reason why the inside remained white may be either (*a*) that it hurried through the temperature just below the eutectic freezing point too fast to allow its cementite to graphitize; or (*b*) that the graphitization was restrained by the pressure put on the interior by the rapid cooling and contraction of the exterior; or (*c*) both these causes may have coöperated. In this particular case his experiments showed that if the slow cooling had not been thus interrupted the interior would have been very gray.

Thus Mr. West confirmed in a very simple and effective way the evidence of Heyn and Bauer that graphitization chiefly occurs immediately below the eutectic freezing point.

The usual occurrence of white cores inside of gray iron pigs so often reported is probably due to pressure. If the outside of the casting becomes firm and rigid, the separation of graphite inside it immediately sets up pressure, because the graphite is so bulky, and the existence of this pressure tends to prevent the further formation of graphite in the parts further in. In most cases this tendency is resisted by the slower cooling of the interior than of the exterior, slow cooling in itself prolonging the opportunity for the graphitization of the cementite.

Thus there is a struggle between slower cooling in the interior which tends to bring the carbon to the state of graphite, and pressure in the interior which tends to prevent it from assuming the form of graphite. In the majority of cases the slower cool-

ing has the mastery, and the interior of the casting is as graphitic as the outside, but in certain cases the effect of pressure has the mastery and restrains the formation of graphite in the interior of the casting. It may often happen that the throwing of a stream of water on the pigs after their outside has turned gray, but before the inside has, or their removal from the sand in cold weather at this point in their cooling, may, as in Mr. West's experiment, lead to an internal chill.

P. Munnoch.[1] It is interesting to find that the phenomenon of internal chill which has puzzled the foundry for so many years has been explained by Mr. West and found to be capable of reproduction. Although the internally chilled castings one sometimes meets in the foundry have not been produced by quenching in water, there is no doubt but that they are due to some rapid cooling effect just at the period when the metal in the center is susceptible to rapid cooling.

In reference to external chilling, the experiments of Mr. West have demonstrated that the chill structure is produced after the metal has solidified. This corroborates the work of E. A. Custer with castings made in chill molds which show no signs of chilled structure provided the castings are removed from the molds immediately after casting.

The placing of a chill in a mold no doubt has a great effect at the beginning of the chilling period, but as the temperature of the face of the chill block becomes heated the chilling effect is reduced considerably before the end of the chilling period, whereas the effect of air chilling will continue equally efficient during the whole of the chilling period. This no doubt accounts for the better results obtained with air cooling by Mr. West.

Paul Kreuzpointner.[2] Summing up the deductions to be drawn from the detailed results of the paper, there are several methods whereby we can obtain a chilled surface in cast iron, the depth of the chill depending upon three factors: (*a*) conductivity of the chilling medium, (*b*) continuity of contact of the chilling medium, and (*c*) quickness of action of the chilling medium, always provided the chemical composition is favorable to obtaining good results.

[1] American Shoe & Foundry Co., Mahwah, N. J.

[2] Secy. American Foundrymen's Association, Altoona, Pa.

When chilling cast iron it is aimed to prevent the changing of the combined carbon into graphitic carbon. The interval of time when this change from combined carbon to graphitic carbon takes place being limited, the success of obtaining a satisfactorily chilled casting depends on the united efficiency of the above three factors; hence the importance of having the proper chilling medium.

If the conductivity of the chilling medium is poor, continuity of contact and quickness of action will not help much. The best conductive media will give inferior results if the contact is broken too soon or if it is applied too slowly. Of course through intensity of application of one of these three factors we may be able to modify, to a certain extent, the unfavorable influences in weakness of the other two factors, but then we enter the realm of chance and haphazard work.

Since the conductivity of a chilling medium is dependent upon its ability to carry away from the casting a maximum amount of heat in the shortest possible time it is obvious that if the chilling medium is a metallic substance, that metal ought to be as dense as possible in structure, because in such a metal there exists the closest contact between the particles, which in turn permits the heat to travel quickly through the mass of the particles or crystals. Therefore wrought iron would seem to be the least adapted to act as a chilling medium because of the presence of the cinder which acts as an obstruction to the rapid traveling of the heat from one particle to another. Theoretically tool steel would seem to be the best chilling medium of the metal class because of its freedom from interposing films of slag or graphite between the crystals, hence maximum conductivity. However, in everyday practice this is out of the question and it is only mentioned in connection with the question of conductivity of chilling media. Thus for everyday practical operations we have to fall back upon cast iron or a suitable stream of cold air, where this is feasible. In the end the problem is always to convey the heat away from the casting quickly enough to prevent change of carbon from one state to the other in order to obtain a desired depth of chill.

James A. Beckett.[1] The immersion test referred to in Pars. 63 and 64, indicating the practicability of continuing chilling

[1] Hoosick Falls, N. Y.

after the metal solidifies, brings out a new idea in connection with this subject and opens a new field for investigation.

Long experience has demonstrated that for surface chilling nothing better has been found, cost and wearing qualities considered, than the cast-iron chiller, which is also a fairly good absorber of heat, and holds its shape well.

Surface chilling is almost entirely a matter of heat transference, but the efficiency of any chilling medium may be increased by the selection of a mixture of irons which have the tendency to a greater increase of combined carbon upon remelting than is found in ordinary foundry irons. Irons of a given analysis made from certain Eastern ores have this tendency to a greater degree than most Southern irons of the same analysis.

The experiment shown in Par. 29 made to determine the effect of warm chillers is in accord with the results of practical experience of long standing. For many years, plow castings have been made by using the chiller for the cope of the mold, in the upper surface of which was formed a receptacle, and into this warm or hot water was poured just before the iron was poured into the mold. The effect of the heat was to reduce the depth of chill in the casting, and by securing uniformity in the temperature of the chiller, to produce a uniform density of grain in the fracture without materially decreasing the strength of the casting. In this way, castings were produced with a close grain and susceptible of a very high polish.

Surface chilling results from the instant withdrawal of heat from the surface of molten iron, and is proportionate only in a limited degree to the thickness of the chiller used, as shown in Par. 31. This is true because the body of the chiller, however thick, will not transfer the heat absorbed by the face of the chiller as rapidly as it is absorbed, hence the resort to water cooling and similar devices. The application of a jet of air to aid in the chilling process is a difficult matter, expensive in the long run, and not likely to come into use except in special cases.

Par. 65 contains a statement of two laws, the second of which is: " Graphitization having once taken place in the crust or body of a hot casting, no sudden cooling can restore the carbon to its original combined form. And only by remelting can it be so transformed as to have a chilled or white iron structure." This law is not as positive in its action as Mr. West assumes, except in

the case of gray iron. White iron used in the malleable iron industry, when subsequently annealed until practically all its carbon is in the graphitic form, may be restored to its original state in which practically all its carbon is combined by heating it to a cherry red and quenching it in cold water.

CARL HERING.[1] This paper is a good illustration of the importance of studying more carefully the phenomenon of the flow of heat through bodies and to other bodies. Like an electric current, a flow of heat encounters a resistance which opposes or checks it more or less, hence when the flow should be rapid, as in the chilling of a casting, this resistance to its flow should be kept as small as possible. The thermal resistance of the cast metal itself is a physical property of the material and is therefore beyond control, but the resistance between it and the chilling material can be controlled, and it is this resistance which is probably so much greater than all the others that it virtually governs the flow of the heat.

The thermal resistance of a joint between two solid bodies, that is, at the surface of a mechanical (not soldered) contact, seems to be relatively very high. This was shown in the case of fire brick in the experiment quoted by the author in Par. 45 and Fig. 15 from an article of the writer's. But with metals this resistance seems to be relatively far greater, as is shown by the fact that an iron bar with a transverse crack part way through it near its end will, when that end is heated to redness, become red up to the crack, while it is still black on the other side. This shows that the heat in flowing to the cold end meets with a very high thermal resistance at the crack, even though it may be only an exceedingly thin one.

In chilling a casting like a car wheel with an iron chiller the heat must traverse the surface of contact and as the cast metal shrinks and the chiller expands, an actual separation must occur, and the very high thermal resistance introduced thereby will no doubt govern the flow of heat and reduce it greatly. This is shown by Mr. West's experiments in which he obtained a better flow of heat when he applied a pressure between the metal chiller and the shrinking casting. It is evident, however, that such a pressure contact cannot possibly be made between two cylindrical surfaces, as in casting car wheels.

[1] Cons. Elec. Engr., 929 Chestnut St., Philadelphia, Pa.

Another and sometimes extremely high thermal resistance exists between a metal and gas.[1] A thin layer of very high resistance seems to be produced on the immediate surface. This is what limits the rate of the flow of heat from the flames to the water tubes of a steam boiler. A postage stamp pasted on the outside of a tin cup in which water is boiled over a flame will not char, showing an enormous drop of temperature through a layer of only a few thousandths of an inch in thickness, which is merely another way of stating that this layer has an extremely high thermal resistance.

This high surface resistance can be destroyed mechanically by carrying away this film of gas, and this is what Mr. West seems to have accomplished by chilling by means of a rapid flow of air in immediate contact with the metal. His results seem to confirm this theory, or if the theory is acknowledged to be correct, his method of air chilling is a rational one, since it enables him to destroy this high contact resistance, which it would be difficult to do in practice between two metallic surfaces, and apparently would be quite impossible when the surfaces are cylindrical as in the case of car wheels.

Another advantage in cooling with rapidly flowing air is that it affords a possibility of controlling the flow of heat from the casting, hence also the depth of chill and its uniformity in successive castings. This would not be easy with metal chillers. Moreover iron chillers are likely to be in actual, good, compressed contact at only a few points, and as the thermal resistance at those points would be considerably lower than at the rest of the surface it is likely that the chilling would be greater at those points, hence not uniform.

A. S. Dowler[2] said in a communication that a method which can increase the depth of chilling after a crust has commenced to expand away from the chiller should permit the use of softer and often stronger grades of iron than are ordinarily permissible in the manufacture of car wheels. Present methods demand a high-chilling iron in order to obtain the desired depth

[1] Flow of Heat Through Contact Surfaces, Met. & Chem. Engrg., January, 1912, p. 40.

St. Louis Car Wheel Company, St. Louis, Mo.

of chill, which, as a rule, is an iron likely to give trouble because of its bordering too closely on a chilled or white structure.

Albert Sauveur.[1] Mr. West's statement to the effect that the chilling of cast iron does not occur entirely while the molten iron is solidifying is undoubtedly true, but the writer thinks that the opposite view is not the accepted idea. That chilling may continue after solidification is not only consistent with what we now understand to be the mechanism of that phenomenon but is demanded by it. By the chilling of cast iron is meant the retention of its carbon in the combined condition, that is, as the iron-carbide Fe_3C or cementite. This carbide must be prevented from dissociating, that is, from breaking up into iron and graphite ($Fe_3C = 3Fe + C$), or, in other words, its "graphitizing" must not be permitted. Cementite, however, represents an unstable condition of carbon while graphite is the stable form of that element. The tendency on the part of cementite to dissociate is the greater the higher the temperature. It is, therefore, maximum during solidification immediately after its formation, but it does not by any means cease once the cast iron has become solid. Indeed it is well known that cementite may be readily decomposed at a temperature as low as 800 deg. cent., that is, some 350 deg. lower than the solidification point of cast iron. To illustrate, cast iron may easily be conceived to solidify so quickly that its carbon is retained entirely in the combined condition, but to cool so slowly afterwards that most, if not all, the carbon passes to the graphitic condition. Clearly the chilling due to rapid solidification was entirely obliterated by subsequent slow cooling. That the chilling, therefore, may continue long after solidification is indeed to be expected.

Bradley Stoughton. Mr. West has again put the foundry industry under obligation for his elucidation of the operation of the chilling of cast iron. It may confidently be expected that his investigations will result in improving the quality of chilled car wheels, cast-iron rolls, plows and other objects made of chilled cast iron.

In two respects the writer must differ with the author of the paper. First, with regard to his statement in Par. 65, "Graphitization having once taken place in the crust or body of a hot

[1] Prof. of Metallurgy and Metallography, Harvard Univ., Cambridge, Mass.

casting, no sudden cooling can restore the carbon to its original combined form, and only by remelting can it be so transformed as to have a chilled or white iron structure." While this statement is no doubt correct under the conditions of his experiments, and in the sense in which he probably intended it, it cannot properly stand in the terms used in his paper, in view of the positive results of G. B. Upton[1] to the contrary, and, in a minor degree, in view of the well-known whitening of some gray and annealed cast irons when they are reheated to temperatures far below their melting point and plunged into water. That is to say, reheating and plunging into water may accomplish what mere plunging into water from a high temperature during cooling after solidification will not accomplish, and what Mr. West says can be accomplished only by remelting.

The writer's second difference relates to the author's statement that it is an accepted idea that chilling occurs entirely while the molten metal is solidifying. It may be an accepted idea among foundrymen who are not familiar with experiments of others in this field, but it has been known and published by more than one observer that chilling may be effected by rapid cooling subsequent to solidification. This correction does not in any way disparage the value of Mr. West's work; on the contrary, the accuracy of his experiments is further confirmed by the fact that they have corroborated the work of others without any previous prejudice in that direction on the part of the investigator.

J. E. Johnson, Jr., said that at the Ashland plant of the Lake Superior Iron & Chemical Company, of which he is manager, they have been conducting a research for a year and a half past on the general subject of the quality of the iron, and particularly on the subject of iron with an internal chill. In charcoal iron and also in coke iron they sometimes found a spot the size of a nickel, chilled perfectly white, in the center of a pig which was perfectly gray on the outside. They do not know the cause, but believe it to be the result of segregation. Analysis showed ½ per cent more carbon in the gray exterior than in the chilled center, but this result has not been confirmed in all cases. The current explanation among scientific people that the central portion is prevented from expanding (which must occur when the

1 The Journal of Physical Chemistry, October 1908, vol. 12.

graphite is evolved), by previous solidification of the crust around it, is not valid as they see it, because all irons are cooled under these conditions, and if one shows an internal chill all should.

They have found that these irons are embraced in what are known as high cleavage irons, which means in plain English, rotten irons of very low strength. This character follows the iron throughout, and in a remelt will show up about the same as in the original iron. Even two or three remeltings will not alter the character of the iron unless something radical is done to change its nature. It will still be spotted iron. As a result, his company never ship this iron to any user who does not alter the fundamental character of the iron before putting it into service.

Their researches have reached the place where they are certain there is virtually no relation between the fundamental quality of the iron and its composition as determined by analysis. There is a certain relation between the percentage of silicon and its chilling quality, but beyond that, with regard to its fundamental characteristics, such as strength, hardness, endurance, etc., an analysis for all the ordinary elements, including carbon, gives no reliable indication. Physical tests, the fracture of the iron, the surface of the pig, its grain and other points must all be considered at least as much as chemical analysis. Even the microscope does not give results that can be depended upon absolutely. His company has been making microscopic examinations of the various irons that they could obtain for over a year, and no relation between the microstructure and the physical qualities has been discovered that they could tie to.

To illustrate the utter unreliability of chemical analysis, Mr. Johnson quoted the records of two cases which were practically identical in all elements, including carbon. With every cast are poured 1¼-in. round test bars. The bars from one of these casts broke at an average of 2725 lb. and those from the other at 4950 lb., or nearly double as much.

The chilled car wheel is bound to be used for years to come and the subject of chilling irons is of enormous importance. Probably 95 per cent of all the freight traffic of the country is carried on chilled cast-iron car wheels. The steel wheel has not, by any means, fulfilled the promise of its advent. The prob-

lem is to make the chilled cast-iron car wheel good, and the writer believed that could be done only by the use of charcoal iron. He had definite and positive knowledge of one foundry, operated by a railroad, under the care of a man who knows his business, where they make wheels for their own use exclusively. Instead of buying junk and treating it with ferro-manganese and other medicines, he buys selected brands of charcoal iron, all tested under his own supervision for their strength and chilling qualities. As a result, he gets a life for his wheels, in service, on one of the great trunk lines of the country, of 95,000 miles, as against a life of from 31,000 to 45,000 for the ordinary wheel made of poorer material.

In concluding, Mr. Johnson read a letter from A. M. Thompson of the Thompson Malleable Iron and Adjustable Clamp Company of Chicago in which he said he had to make some rolls which called for a chilled iron. He made the usual mixture of steel and ordinary iron, and test bars 2 in. by 4 in. in size showed $\frac{5}{8}$ in. chill. Reports from the machine shop and rolling mill indicated that the surface was soft. Sulphur was added to the liquid iron and the rolls still continued soft. Some of them were taken out of the chill as soon as practicable after casting and cooled in water with no perceptible difference in hardening. A charcoal chilling iron was then purchased which proved entirely satisfactory. Previous to that he had believed in common with many foundrymen that where the fracture was white the carbon was combined, no matter what iron was used, and the limit of hardness had been reached.

H. M. Lane reported an incident where a foundryman was pouring stove plate. He used a ladle which had been dampened, after drying it slightly on the surface so that only just enough steam would form to cause the iron to boil. He poured one plate from that ladle and another from a perfectly dry ladle. When the plates were broken the first one was found to be chilled on the inside and the other was not. The steam boiling through the iron had caused some change either physical or chemical which produced an internal chill, with $\frac{1}{32}$ to $\frac{1}{16}$ in. of good gray iron on the outside, the inside being hard and white so it could not be drilled. The same thing often occurs when the bed of the cupola has not been dried out thoroughly. It causes the iron to

boil at the start and the first 100 or 200 lb. will show internal white marks.

With regard to the addition of sulphur, which has been mentioned, a patent process has been introduced into this country at least five times involving the use of sulphur in steel making. It comes from Germany and consists in adding iron pyrites to a malleable charge to introduce sulphur into the iron. Two-inch and two and one-half-inch sections are chilled all the way through and when they are treated in the annealing furnace they produce a fair grade of malleable.

A. E. Outerbridge, Jr.[1] wrote that he believed the author's discovery of a method of producing at will an inside chill in cast iron to be new and, apart from any application that it may have, important as tending to solve a mystery that has long puzzled metallurgists. The fact that a chilled face of a casting can be rendered harder by a cold air blast is also a valuable contribution to the metallurgical art.

At one time the writer had several car loads of imported pig iron showing large open dark gray grains and an internal chill in the center. This iron was quite high in manganese and he attributed the internal chill to that element. When used in the regular car-wheel mixture, no tendency to cause an internal chill was observed.

He can fully substantiate the statement as to the effect of warming the chiller, for he has records showing reduction of chill in every case.

The Author. Mr. Beckett states that "The application of a jet of air to aid in a chilling process is a difficult matter, which will prove expensive in the long run and is not likely to come into use except in special cases." Time only can tell whether his modification of "except in special cases" will hold. The author has patented processes whereby varying conditions can be met with by the use of suction as well as with air pressure. One of these is described in Par. 59 and illustrated in Figs. 9 and 10, showing that with air treatment simple methods can be used to cool and deepen or harden a chill. The illustrations of the paper and specimens exhibited at the meeting forcibly demonstrate the effects of air treatment.

[1] Metallurgist, Wm. Sellers & Co., Inc., Phila., Pa.

In commenting on Par. 65, which says "Graphitization having once taken place in the crust or hot body of a casting, no sudden cooling can restore the carbon to its original form, and only by remelting can it be so transformed as to have a chill or white iron structure," Mr. Beckett gives valuable information but little known. However, it is to be understood that the author had in mind the possibility of changing the state of the carbon at the time of making a casting. This also applies to one of the points raised by Mr. Stoughton.

Mr. Munnoch's discussion states that the author's work in demonstrating that the chill structure is produced after the metal has solidified corroborates the work of E. A. Custer. The paper states that the chilling of iron must be accomplished prior to the formation of graphite, and that it has always been thought that in chilling iron the action ceases the moment the metal has solidified. This does not imply that an iron is not chilled until after the metal has solidified. The point desired to be emphasized is that chilling can be continued up to a period of 20 or 30 seconds after the moment of solidification, and Mr. Munnoch has evidently misunderstood the meaning. The writer believes that Mr. Custer's hypothesis, while having some relation to the above question, is not strictly within the lines of research covered by the paper in question.

No. 1351

TESTS OF CHILLABLE IRONS

By Thos. D. West, Cleveland, Ohio

Member of the Society

The paper presents an original series of tests of chillable irons, extending from September 1911 to the close of February 1912.

2 Before proceeding with the tests it was desired to find what size of bar should be used with different grades of iron to produce a bar of all gray iron and at the same time a companion bar that could be wholly chilled, or of a white iron, both poured from the same basin or ladle. Sets Nos. 1 and 2 show that bars $1\frac{5}{8}$ in. in diameter are suitable for various grades of chillable metal having its silicon ranging from 0.90 to 1.20. The balance of the sets show $2\frac{1}{4}$ in. diameter bars to be suitable for many grades having a range of 0.50 to 0.90 per cent silicon. It is to be understood that in either of the above the constituents, other than silicon, are generally the same as used in the making of such castings as chilled car wheels and rolls. In some cases the larger bars may be used for higher silicon metal, this depending chiefly upon the metal being high in sulphur and no ferro-manganese being used.

3 While the round bar is advocated for a standard, it is to be understood that there are cases of experimental work such as presented in this paper, where square bars $1\frac{1}{2}$ in. or over may be advisable, but for ordinary practice to obtain comparisons in mixtures, grades of metal, etc., the round bars are to be preferred.

4 For molding the square bars three flasks were constructed, each being adapted to cast three or four bars. These bars were 2 in. sq. by 24 in. long. For casting the round bars, two chiller molds having a bore of $1\frac{5}{8}$ in., later on bored out to $2\frac{5}{16}$ in., were used in connection with two pipe sand molds. The designs for both are seen in Figs. 1 to 6 in the paper, A Suggested System of Test Bars for Chillable Irons, to be presented by the writer before the Sixth

Presented at the Spring Meeting, Cleveland, 1912, of The American Society of Mechanical Engineers.

Congress of the International Association for Testing Materials in September 1912. The three sets of flasks were often necessary, as in the case of making bars to test the relative effects of vanadium and titanium as alloys.

METHODS FOR OBTAINING AND ALLOYING METALS TO TEST THEIR EFFICIENCY

5 In casting sets for these tests, the bars were poured with the regular metal from a reservoir ladle under the cupola spout, with a capacity of about 7 tons, and carried to the molds in a "bull ladle" which held about 250 lb. Twelve ounces of ferro-manganese was thrown into the ladle, in order to secure the same composition as used for car wheels where 2 to $2\frac{1}{2}$ lb. of ferro-manganese is added to every 700 or 800 lb. of metal.

6 After the bars were poured of this regular wheel metal the bull

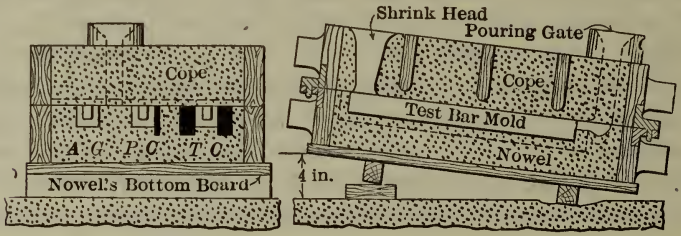

FIG. 1 SECTIONAL VIEWS OF MOLD FOR MAKING THE SQUARE TEST BARS

ladle was again filled as often as needed and vanadium or titanium, or both together, added according to the tests to be made, along with the 12 oz. of manganese. The ladle was allowed to stand for three or four minutes to permit the alloys to melt thoroughly and mix with the metal. A $\frac{1}{2}$-in. rod was used to agitate the metal to help bring any oxides created by the alloys to its surface. The two, three or more bull ladles of metal required for a set were taken from the reservoir ladle before any additional tap of metal was run into it from the cupola.

METHOD OF CASTING AND CHILLING THE ALLOYED METAL

7 In pouring either of the above alloyed metals with the regular metal to obtain a set of square bars, a set of round bars would be poured with the same ladle.

8 In casting the square test bars, some had a 2 in. by $2\frac{1}{2}$ in. chiller on two sides, as illustrated at *TC* in the end view of Fig. 1, so as thoroughly to chill the bar to make it all white iron. Others

had a wrought plate ¼ or ½ in. thick as desired, on one side of the bar only, as at *PC*, in order only partly to chill one side of the bar. Bars to be free of chill, were surrounded with sand as seen at *AG*. In some cases two totally chilled bars and one all gray bar might be cast in one flask. Again, two partially chilled, one totally chilled, and one all gray bar might be cast in one flask. This order could be changed in providing for three to four bars being made in a flask.

9 The character of the chill, or grain of iron, is given in the tables under the heading Fracture. Should an all sand molded bar show

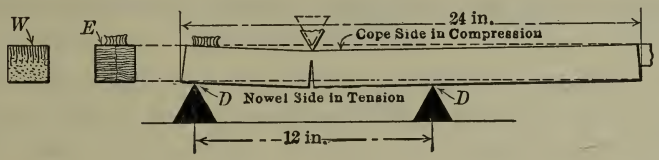

FIG. 2 END AND SIDE VIEWS OF A TEST BAR MADE IN MOLD, FIG. 1
W shows Gray Side in Tension and Strong Position; *E* shows Horizontal Plane of Chill Crystal in Tension and Weak Position

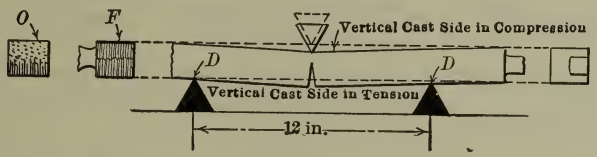

FIG. 3 SECOND TEST ON A 2-IN. SQ. BAR
O shows Chilled Side in Tension and Weak Position; *F* shows Vertical Plane of Chilled Crystal in Tension and Strong Position

a slightly mottled grain in its fracture instead of being all gray, the words slightly mottled are inserted. Should the fracture be strongly mottled the notation is the word " deeply " in the place of " strongly." In cases where the depth of a partly chilled bar was measured the thickness of the chill is given in connection with the statement that one side was chilled.

10 To indicate that one of the vertically cast sides, also the cope surface, or the nowel face of a bar is in tension when testing a bar, the words nowel, cope and side are placed on the line with the respective bar in the column headed Tension. This will be better understood by referring to Figs. 2 and 3 where the side and end view of the bars are shown in two ways of being tested. In the case of round, as well as square bars that are cast on end, as were Nos. 84

and 88, it makes no difference which way they are placed on the distance supports *D*, Figs. 2 and 3, when being tested.

11 The width and depth as well as the diameter of all the tested bars are given in Tables 3–10 to permit a checking of the modulus of rupture column; also present data for other formulae for computing variations in the size of the bars shown. It is to be understood that the records of all tests given are of solid bars and complete fractures; as should there have been a slight flow in any of those tested it would have been in the compression face of the bar where it could have no effect in reducing its strength.

METHODS USED FOR TRANSVERSE TESTING AND PRACTICABILITY OF DROP TESTS

12 All transverse tests on both round and square bars, Tables 5 to 9, except tests 59 and 77, were made with a 12-in. span, as seen in Figs. 2 and 3. Tests 59 and 77 were made with a

TABLE 1 RANGE OF CHEMICAL ANALYSIS FOR CHILLED CAR WHEELS

Silicon	Sulphur	Manganese	Phosphorus	Combined Carbon	Graphitic Carbon
0.55 to 0.65	0.100 to 0.150	0.55 to 0.70	0.270 to 0.320	0.60 to 0.70	2.70 to 2.90

10-in. span. All drop tests of Tables 5 to 9 were made with a 12-in. span and all drop tests for Table 10 were made with an 8-in. span.

13 The drop tests were made with a 25-lb. weight having the first drop at a height of 6 in. and raised 6 in. higher for every blow until breaking the bar. If, for example, 6 is the number in the drop column of the tables, it means that the weight dropped once at each of the respective heights, 6 in., 12 in., 18 in., 24 in., 30 in., and 36 in., before breaking the bar. The bars for the drop tests Tables 5 to 9 were obtained by taking the long end of the 24-in. long square bars after they had stood one transverse test. This is why two kinds of tests with the same bar are given under the same test number.

14 A comparison of the drop tests with the transverse tests shows that where a bar is strong transversely it generally shows a relatively high strength under the drop test. This was also true in the case of nearly all of the round bars having a 6-in. span, but these tests are not given herein.

15 For castings that are subject to shock or sudden impact, such as car wheels, rolls, shears, dies, etc., the drop test should grow in favor. The apparatus costs little and the time required is less than that needed in any other method of testing.

TREATMENT AND ANALYSIS OF REGULAR CAR WHEEL AND ALLOYED METALS

16 No special complete analysis is given of the respective sets

TABLE 2 ANALYSIS OF SETS ALLOYED WITH VANADIUM AND TITANIUM

Set No.	Oz. of Vanadium in 225 Lb. of Metal	Oz. of Titanium in 225 Lb. of Metal	Per Cent of Vanadium in Test Bars	Per Cent of Titanium in Test Bars	Per Cent of Manganese in Test Bars
9	10	..	0.06	....	0.63
11	..	10	....	0.02	0.72
13	22	..	0.12	....	0.64
15	..	22	....	0.07	0.60
18	5	10	0.04	0.03	0.69
19	8	15	0.05	0.05	0.68

TABLE 3 TRANSVERSE TESTS 1¼ IN. SQ. AND 1⅝ IN. ROUND BARS AND CRYSTALLIZATION

Test No.	Fracture	Width	Depth	Tension	Maximum Load	Deflection	Modulus	Set No.
1	All white	1.73	1.72	Nowel	5140	0.016	18070	1
2	All white	1.68	1.69	Side	13160	0.028	49380	1
3	Slightly mottled	1.75	1.76	Side	15300	0.060	50820	1
4	All white	1.59	diameter	. . .	4780	0.023	36280	1
5	Slightly mottled	1.60	diameter	....	7340	0.065	54670	1
6	All white	1.69	1.69	Nowel	7580	0.028	28260	2
7	All white	1.71	1.71	Side	15280	0.042	55010	2
8	Strongly mottled	1.72	1.79	Side	11940	0.038	39000	2
9	All white	1.60	diameter	....	5820	0.032	43350	2
10	Almost white	1.56	diameter	....	5540	0.040	44520	2
11	All white	1.59	diameter	....	3080	0.021	23370	3
12	Strongly mottled	1.58	diameter	....	7760	0.073	60030	3

other than that in Table 1. This is owing to the fact that car-wheel mixtures or analyses, from one foundry at least, vary but little. Table 2 gives the vanadium and titanium constituents of the test bars. Analyses of metal taken directly from the reservoir ladle before the ferro-manganese was added showed this metal to contain around 0.40 manganese. The drillings for regular car-wheel metal

analyses are taken from blocks about 2 in. by $2\frac{1}{2}$ in. by 8 in. cast in all sand so as to leave a gray body in the metal, which will agree closely with the gray between the plate and body back of the chill of a car wheel.

SENSITIVE CONDITIONS REQUIRING CONSIDERATION AND CONTROL IN TESTING CHILLABLE IRONS

17 Table 3 shows that bars $1\frac{3}{4}$ in. square and $1\frac{5}{8}$ in. round are too small for regular car wheel metal. This is seen by the all-sand bars 3, 5, 8, 10 and 12, showing a fracture a little too strongly mottled instead of a fair gray structure. These sizes of bars are recommended only for chillable irons ranging from 0.90 to 1.20 per cent in silicon for use in general practice.

18 The foregoing is a general statement requiring modifications in some instances. A comparison of the strength of bars 4, 5, 11 and 12 with 9 and 10 shows the cases in which strongly mottled iron of the same metal, also iron tending to all white, as No. 10, will give exceptional strength. Many other round bars $1\frac{5}{8}$ in. in diameter were made, but broken with the sledge to check these fractures. The two odd bars 11 and 12 are shown chiefly to demonstrate the distinction they display.

19 When a bar is of such a size that it is sensitive to assume a mottled form, it is very likely to go further and become almost white, and with the same iron, temperature of metal and character of mold, there is as much chance to obtain a strength of 7760 lb. as 5540 lb., as seen for bars 10 and 12. But this sensitive condition must be avoided the better to make comparisons between an all-chilled bar and an all sand cast one of the same size, form, and metal. To do this, it is necessary to have the sand cast bars sufficiently large to prevent their taking a mottled form, and still not so large as to prevent their being absolutely chilled, or all white iron to their very center, when cast in an all-iron mold or chiller of the same diameter. Much experimenting may often be necessary to learn to know the best size to adopt for making a comparison between the white and gray of special chillable irons. It will be seen by the tables having the $2\frac{1}{4}$-in. round bars, that even with this increase over the $1\frac{5}{8}$ in. in diameter, some of the larger bars show a slightly mottled fracture with the silicon around 0.60. This could have been largely avoided by baking or drying the sand molds, as those used in the tests shown were all of green sand. To increase the size of the round bar would assist this feature, and such could be done to the extent of having it

$2\frac{3}{4}$ in. and possibly 3 in. in diameter and still produce a perfect, all-chilled bar for a companion to an all sand cast or gray one having the silicon around 0.60.

20 It is important therefore to describe the structure of fractures when recording tests of chillable iron and in making comparisons with all-chilled and gray bars or otherwise. It all shows that in some cases, it may be necessary to experiment in order to obtain the diameter best suited to give a knowledge of the relative strength of

TABLE 4 TRANSVERSE TESTS OF VARIABLE DEPTHS OF CHILL WITH LOW CHILLING IRONS

Test No.	Fracture and Per Cent of Chill	Width	Depth	Load	Deflection	Modulus	Set No.
13	Not chilled; clear gray iron	1.01	1.51	5615	0.098	43900	4
14	Chilled one side; 10% white	1.00	1.50	4360	0.085	34900	4
15	Chilled two sides; 25% white	0.96	1.44	3270	0.056	29600	4
16	Not chilled; clear gray iron	0.98	1.46	5175	0.105	44500	5
17	Chilled one side; 15% white	1.03	1.48	4220	0.069	33700	5
18	Chilled two sides; 20% white	1.02	1.45	4390	0.079	36900	5
19	Not chilled; clear gray iron	0.97	1.47	5000	0.103	43000	6
20	Chilled one side; 1% white	1.00	1.47	5610	0.108	46750	6
21	Chilled two sides; 100% white	1.01	1.47	6400	0.080	52800	6
22	Not chilled; clear gray iron	1.00	1.46	4350	0.157	36700	7
23	Chilled one side; 5% white	0.99	1.43	4550	0.159	40500	7
24	Chilled one side; 20% white	1.06	1.50	4200	0.137	31700	7
25	Chilled two sides; 50% white	1.02	1.50	2300	0.110	18050	7

the white and gray of chillable irons. This does not prevent the adoption of a standard to be used the world over[1] for tests for chillable irons. All that is necessary is to state the structure of the fracture, diameter of the bar used, per cent of silicon, and possibly other constituents, should they vary much from those given in Table 1.

ERRATIC EFFECTS OF CHILLED CRYSTALS AND INTERLACING OF THE MOTTLED AND GRAY WITH THEM

21 Table 4 presents a few of the many tests made with chillable metal, having about the following composition: silicon 2.0, sulphur 0.06, phosphorus 0.04, manganese 0.30. The bars were made in a

[1] Suggested by the author in A Suggested System of Test Bars for Chillable Irons, presented at the Sixth Congress of the International Association for Testing Materials, held September, 1912.

TABLE 5 BARS 26 TO 30 HAD 12 OZ. MANGANESE; 31 TO 34, 12 OZ. MANGANESE AND 10 OZ. VANADIUM IN 225 LB. OF METAL

Test No.	Fracture and Per Cent of Chill	Width	Depth	Tension	Load	Deflection	Modulus	Drop	Set No.
R 26	Chilled both sides. All white iron	2.04	2.00	nowel	15150	0.030	33420	4	8
R 27	Chilled ½ in. one side	2.10	2.05	nowel	14330	0.036	29230	6	8
R 28	All gray iron	2.12	2.08	nowel	23860	0.070	46820	9	8
R 29	All gray iron	2.31	diameter	....	25750	0.095	63710	..	8
R 30	Chilled. All white iron	2.21	diameter	....	14620	0.030	41330	..	8
V 31	Chilled two sides. All white iron	2.08	1.97	nowel	16100	0.030	35900	4	9
V 32	Chilled ⅝ in. one side	2.05	1.99	nowel	15370	0.035	34080	5	9
V 33	All gray iron	2.08	2.00	nowel	22020	0.070	47640	9	9
V 34	All gray iron	2.32	diameter	....	23430	0.070	57300	..	9

Set *R* poured with regular iron. Set *V* poured with vanadium in regular iron. Set *T* poured with titanium in regular iron.

TABLE 6 BARS 33 TO 39 HAD 12 OZ. MANGANESE; BARS 40 TO 44, 12 OZ. MANGANESE AND 10 OZ. TITANIUM IN 225 LB. OF METAL .

Test No.	Fracture and Per Cent of Chill	Width	Depth	Tension	Load	Deflection	Modulus	Drop	Set No.
R 35	Chilled both sides. All white	2.14	1.98	nowel	22420	....	48108	..	10
R 36	Chilled ⅛ in. one side	2.03	2.07	nowel	18990	0.037	39170	3	10
R 37	All gray iron	2.12	2.04	nowel	25740	0.075	54700	11	10
R 38	All gray iron	2.27	diameter	....	25520	0.077	66570	..	10
R 39	Chilled. All white iron	2.22	diameter	....	13440	0.024	37477	..	10
T 40	Chilled both sides. All white	2.02	1.97	nowel	16100	0.031	36960	4	11
T 41	Chilled ⅜ in. one side	2.07	2.00	nowel	15130	0.032	33280	5	11
T 42	All gray iron	2.06	1.99	nowel	21870	0.074	48250	9	11
T 43	All gray iron	2.32	diameter	....	23400	0.059	57180	..	11
T 44	Chilled. All white iron	2.22	diameter	....	17110	0.028	47711	..	11

TABLE 7 BARS 45 TO 49 HAD 12 OZ. MANGANESE; BARS 50 TO 54, 12 OZ. MANGANESE AND 22 OZ. VANADIUM IN 225 LB. OF METAL

Test No.	Fracture and Per Cent of Chill	Width	Depth	Tension	Load	Deflection	Modulus	Drop	Set No.
R 45	Chilled both sides. All white	1.90	2.02	nowel	16460	0.030	38220	5	12
R 46	Chilled ⅜ in. one side	2.10	2.00	nowel	14380	0.035	30810	7	12
R 47	All gray iron	2.08	2.01	nowel	23790	0.090	50960	11	12
R 48	All gray iron	2.29	diameter	....	23910	0.078	60730	..	12
49	Chilled. All white iron	2.22	diameter	....	15610	0.022	43520	..	12
V 50	Chilled both sides. All white	2.00	1.98	nowel	16030	0.035	36800	4	13
V 51	Chilled ⅛ in. one side	2.10	2.02	nowel	14090	0.030	29600	3	13
V 52	Gray, slightly mottled	2.05	2.00	nowel	24800	0.065	54440	7	13
V 53	Gray, slightly mottled	2.27	diameter	....	25580	0.060	66710	..	13
V 54	Chilled. All white iron	2.20	diameter	....	17890	0.025	51250	..	13

TABLE 8 BARS 55 TO 59 HAD 12 OZ. MANGANESE; BARS 60 TO 64, 12 OZ. MANGANESE AND 22 OZ. TITANIUM IN 225 LB. OF METAL, WHILE BARS 65 TO 68 WERE FREE FROM MANGANESE AND OTHER ALLOYS

Test No.	Fracture and Per Cent of Chill	Width	Depth	Tension	Load	Deflection	Modulus	Drop	Set No.
R 55	Chilled both sides. All white	2.00	2.00	nowel	15980	0.030	35960	4	14
R 56	Chilled 3/8 in. one side. Balance mottled	2.00	2.07	nowel	18140	0.038	38100	4	14
R 57	Gray ron. Corners chilled	2.02	2.00	nowel	22720	0.070	50610	7	14
R 58	Gray iron, slightly mottled	2.25	diameter	....	26000	0.075	69640	..	14
59	Chilled. All white	2.24	diameter	....	13830	0.017	33966	..	14
T 60	Chilled both sides. All white	2.00	1.98	nowel	13200	0.025	30300	4	15
T 61	Chilled 1/2 in. one side. Balance mottled	2.04	2.03	nowel	16840	0.035	36060	4	15
T 62	Gray, corners slightly chilled	2.05	2.00	nowel	22080	0.075	48470	10	15
T 63	Gray iron	2.26	diameter	....	25810	0.08	68230	..	15
T 64	Chilled. All white iron	2.22	....	....	16070	0.05	44810	..	15
S 65	Chilled both sides. All white	2.04	2.04	nowel	11400	0.020	24170	3	16
S 66	Chilled 5/8 in. one side. Deeply mottled	2.06	2.04	nowel	10860	0.030	22800	3	16
S 67	Gray, corners mottled	2.04	2.00	nowel	17870	0.050	39420	5	16
S 68	Gray iron, mottled	2.25	diameter	....	24180	0.065	64204	..	16

Set *S* poured with spurious metal containing no ferro-manganese.

TABLE 9 BARS 69 TO 72 HAD 12 OZ. MANGANESE; BARS 73 TO 77, 10 OZ. TITANIUM AND 5 OZ. VANADIUM; BARS 78 TO 82, 12 OZ. MANGANESE, 15 OZ. TITANIUM AND 8 OZ. VANADIUM IN 225 LB. OF METAL

Test No.	Fracture and Per Cent of Chill	Width	Depth	Tension	Load	Deflection	Modulus	Drop	Set No.
R 69	Chilled both sides. All white	1.98	2.02	nowel	18480	0.035	41170	5	17
R 70	Chilled 1/4 in. one side. Balance mottled	2.04	2.50	nowel	16760	0.055	23660	6	17
R 71	All gray iron	2.05	2.05	nowel	24390	0.085	51020	11	17
R 72	All gray iron	2.27	diameter	....	24030	0.065	62660	..	17
*TV*73	Chilled both sides. All white	2.02	2.00	nowel	12330	0.025	27470	..	18
*TV*74	Chilled 1/8 in. one side. Balance mottled	2.05	2.07	nowel	18340	0.040	37580	6	18
*TV*75	All gray iron	2.10	2.04	nowel	24980	0.065	51390	11	18
*TV*76	All gray iron	2.29	diameter	....	22070	0.060	56060	..	18
*TV*77	Chilled. All white iron	2.21	diameter	....	16370	0.029	38560	..	18
*TV*78	Chilled both sides. All white	1.96	2.03	nowel	16250	0.030	36210	4	19
*TV*79	Chilled 1/4 in. one side. Balance mottled	2.00	2.04	nowel	14580	0.040	31530	5	19
*TV*80	All gray iron	2.04	2.00	nowel	22240	0.065	49060	8	19
*TV*81	All gray iron	2.29	diameter	....	23470	0.060	59620	..	19
*TV*82	Chilled. All white iron	2.20	diameter	....	18700	0.030	53570	..	19

converter steel foundry and tested by Prof. John H. Nelson; all others were tested by Mr. H. E. Smith. In testing this set, the load was applied in the deep direction of the bars which were all cast on end. Tests 13 to 25 illustrate the erratic qualities of partly chilled bodies, accounted for by the interlacing of the white with the gray and the depth of mottled iron back of the chilled body, all causing greater or less internal strains that can often be greatly reduced by annealing.

22 In partly chilled sections, the temperature of the chiller to chill the iron, and of the metal to pour the mold, and the degree of dampness in the sand, have an effect both on the depth of chill, and on the structure of the metal for a considerable distance beyond the place where the white ceases. These are all factors difficult to control in regular practice, but the more that is known concerning them, the better will be the design, make and use of the castings. The variable hardness of mottled and gray bodies, interlacing with the white iron of chilled bodies, is displayed by the hardness tests, Table 12.

TRANSVERSE AND DROP TESTS OF GRAY AND CHILLED BARS ALLOYED WITH VANADIUM AND TITANIUM

23 Tables 5 to 9 present an original series of tests comprising the following features:

a Comparison of strength, deflection, chill and contraction, in all-chilled, partly-chilled, and gray bars of the same metal.

b Comparison of square and round bars to emphasize the utility of the latter for a standard.

c Comparisons of transverse and drop tests to show their conformity, and practicability of the latter.

d Comparisons of hardness created by the rate in cooling, giving chilled, mottled and gray fractures in the same metal.

e Effects of ferro-manganese, vanadium, and titanium in the same metal and size of section, when of a chilled, mottled and gray structure.

24 Results of the above comparisons in connection with those to be derived from a study of the tables are given throughout the paper.

NOTABLE DIFFERENCE IN THE STRENGTH OF THE CHILLED AND GRAY SIDES OF A PARTLY CHILLED CASTING

25 It will be seen from tests in Table 10 that when the chilled face is in tension, as with Tests 92, 94 and 96, the casting is much

weaker than when the gray or mottled body is in tension, as with Tests 91, 93 and 95 of Set 22. This is a quality having heretofore received very little, if any, thought. When fully considered it will be seen to be of great importance in the making and use of different lines of castings. The reliability of this set of tests will be realized when it is understood that the respective companion tests having the chilled side in compression and tension were made

TABLE 10 TRANSVERSE AND DROP TESTS OF ONE SIDE CHILLED BARS ALTERNATED TO BE IN COMPRESSION AND TENSION

Test No.	Fracture and Per Cent of Chill	Width	Depth	Load	Deflection	Modulus	Drop	No.
83	Gray iron, slightly mottled	2.09	2.06	21070	0.045	42760	7	20
84	Chilled four sides. All white	2.12	2.18	23380	0.040	41770	15	20
85	Chilled 5/16 in. one side. Balance mottled. Gray side in tension	2.06	2.05	21590	0.030	43435	9	20
86	Chilled 5/16 in. one side. Balance mottled. Chilled side in tension	2.07	2.03	20430	0.035	43111	7	20
87	Gray iron, slightly mottled	2.11	2.07	19420	0.045	38664	8	21
88	Chilled four sides. All white	2.06	2.20	22450	0.020	40532	15	21
89	Chilled 11/16 in. one side. Balance mottled. Gray side in tension	2.10	2.04	19880	0.050	40947	8	21
90	Chilled 11/16 in. one side. Balance mottled. Chilled side in tension.	2.06	2.06	13660	0.030	28132	6	21
91	Chilled 1/4 in. one side; 3/4 in. mottled. Gray side in tension	2.11	2.15	23760	0.070	43861	14	22
92	Chilled 1/4 in. one side; 3/4 in. mottled. Chilled side in tension	2.13	2.06	17260	0.035	34373	..	22
93	Chilled 1/4 in. one side; 3/4 in. mottled. Gray side in tension.	2.12	2.10	21760	0.055	41914	7	22
94	Chilled 1/4 in. one side; 3/4 in. mottled. Chilled side in tension.	2.13	2.06	13190	0.025	26378	..	22
95	Chilled 1/2 in. one side; 1 in. mottled. Gray side in tension.	2.11	2.03	20350	0.055	42127	5	22
96	Chilled 1/2 in. one side; 1 in. mottled. Chilled side in tension.	2.13	2.13	15720	0.030	29281	..	22

with the same bar, by the method shown in Figs. 2 and 3. After making two transverse tests of the same bar, there was sufficient remaining for a drop test having an 8-in. span. A few of these are Tests 83 to 96. Bars 83 to 90 were cast on end, while bars 91 to 96 were cast flat, as shown in Fig. 1, and chilled on one side only to give two of this form for one set. In Fig. 6 is seen a full set of the square bars cast on end, in which *M* is the all chilled bar, *N* the chilled side, and *O* the gray side of the partly chilled bars, while *P* is the all sand cast bar. The position of the chilled face

in the testing is shown at *W* for both the cast on end and cast flat bars when upward, and at *O* when downward, seen on the left of Figs. 2 and 3.

26 Another feature is the great difference between the strength of chilled iron when the lines of crystallization stand vertical to the load, and when they are turned horizontal to it. In Table 3, Tests 1, 2, 6 and 7 show a difference of about 61 per cent for the first two bars, and about 51 per cent for the second two. The lines of crystallization are seen in Figs. 2 and 3, where *E* is the weakest and *F* the strongest position of the two-sided bar. These qualities were original-

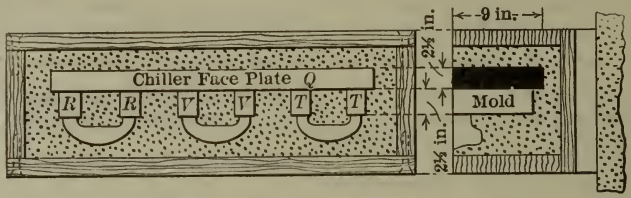

Fig. 4 Open Sand Mold for making Comparative Chilling Tests

ly discovered by Asa W. Whitney, and are presented here to give data in keeping with the original tests of this paper.

CRITICISM ON AND CHILLING EFFECTS OF VANADIUM AND TITANIUM

27 William H. Hatfield in a paper On The Influence of Vanadium upon Cast Iron [1] presented at the March 1911 meeting of the Iron and Steel Institute, stated, "There is considerable disagreement as to the influence of vanadium." Expressions of this character had much weight in the taking of extra precautions when testing these alloys, in the belief that the results might settle some of the disputed points.

28 Information on the chilling qualities of the alloys is given in Pars. 29 to 31 and in the various sets of Tables 5 to 9, as Tests 27 and 32. It required but a few tests to show that the difference in the pouring temperature of the metals, due to the cooling effect produced in melting the alloys, was such as to make the depth of chill shown in Tables 5 to 9 an uncertain factor for these tests. This is more fully realized when the fact is considered that "hot" metal will chill deeper than "dull" metal.

29 To obtain more favorable conditions for rapid pouring and less travel of metal than was offered by the mold, Fig. 1, cupola chill

[1] The Journal, Iron & Steel Institute, vol. 83, no. 1, p. 318.

blocks were made after the plan and end views in Fig. 4. The pair of chilled blocks R were poured with the regular car wheel metal, cooled down to nearly the same temperature as that for pouring the chill block molds V and T having the vanadium and titanium alloys in their respective ladles. This method emphatically demonstrated that the vanadium increased the depth of chill, or held the carbon more in its combined form, while the titanium operated in the opposite direction. Numerous tests were made following this plan, some of

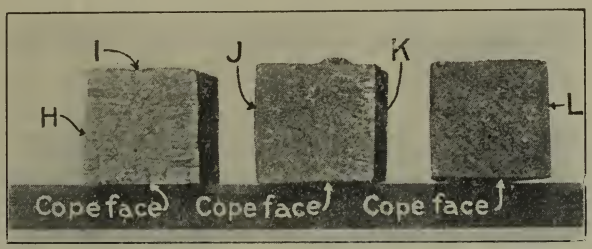

FIG. 5 SET OF ALL CHILLED, PARTLY CHILLED AND ALL SAND SQUARE BARS CAST FLAT

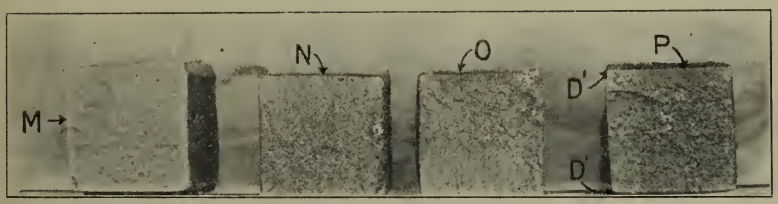

FIG. 6 SET OF ALL CHILLED, PARTLY CHILLED AND ALL SAND SQUARE BARS CAST ON END

which had a $\frac{1}{2}$-in. chiller face plate in place of the $2\frac{1}{2}$-in. plate Q, and all of them were effective in the same direction as to their respective results.

30 One test with the three sets of molds, Fig. 4, gave a difference in the thickness of chill as seen at X, Y and Z, Fig. 7. The Set R was poured with the regular iron; the Set V with the vanadium and the Set T with the titanium alloyed metal, there being 1 lb. of each alloy in about 175 lb. of metal.

31 Further experiments having 2 lb. of titanium in the 175 lb. of metal gave a thickness of chill seen at V and W, Fig. 7, of 1 in. in the regular metal and $\frac{1}{2}$ in. in the alloyed metal, as marked in

the cuts. In the belief that by increasing the amount of titanium the chill might be wholly prevented, 4 lb. was put into the metal for two tests. This did not act as effectively as the 2 lb. and showed that iron could not be prevented from chilling beyond a certain limit by its use.

EFFECTS OF VANADIUM, TITANIUM AND OTHER FACTORS ON CONTRACTION

32 Tests to obtain contraction were made with both round and square bars and are given in Table 11. The $2\frac{1}{4}$-in. round bars show the contraction for a length of 12 in. and the square bars for 22 in. The ratios for contraction of the bars cast on end agree closely with those of the bars cast flat. The regular irons are fairly uniform in

TABLE 11 CONTRACTION OF ROUND BARS CAST ON END AND SQUARE CAST FLAT

Set No.	Round Bars, All Chilled	Round Bars, All Gray	Square Bars, All Chilled	Square Bars, Part Chilled	Square Bars, All Gray
R 12	0.22	0.12	0.47	0.28	0.26
V 13	0.23	0.13	0.48	0.30	0.27
R 14	0.22	0.12	0.47	0.28	0.26
T 15	0.21	0.11	0.43	0.29	0.25
S 16	...	...	0.50	0.36	0.32
R 17	...	...	0.48	0.31	0.28
TV 18	0.22	0.12	0.47	0.29	0.26
TV 19	0.23	0.12	0.48	0.30	0.27

their contraction. The vanadium bars show a greater contraction than those containing titanium, the latter having the least of any of the metals. The spurious metal of Set 16 having no ferro-manganese in it, shows the greatest contraction. The most radical difference exists between the all-chilled and all gray bars.

EFFECTS OF VANADIUM AND TITANIUM ON STRENGTH

33 In making deductions of the relative strength for the alloy mixtures, etc., the round bars were selected chiefly on account of their uniform structure and greater uniformity of results. Fig. 8 is a good illustration of the uniformity of the metal as it comes in round bars rather than in square ones. The gray round bar *A'* shows a much better uniformity of grain structure than exists in the square bars *L* and *P*, Figs. 5 and 6. These last present irregular patches of grains embodying every structure from white at the corners *D'*, interlaced with mottled, leading to a gray center, sensitive to change with the least variation in the dampness or character of the sand, hardness in ramming and temperature of pouring metal. This irregularity

of structure is likewise apparent in the all white square bars *H* and *M*, Figs. 5 and 6, when compared to that seen at *B'*, Fig. 8.

34 The titanium bars show an increase of strength over the regular bars of 27 per cent in the white iron with bars 39 and 44, Table 6, and 32 per cent in the white iron with bars 59 and 64, Table 8.

35 The vanadium shows an increase of strength of 9 per cent in the gray iron, with bars 48 and 53, and 17 per cent in the white with bars 49 and 54, both of Table 7.

36 The spurious bars which have no ferro-manganese or other alloys in them, show a decrease of strength of 7 per cent in the gray

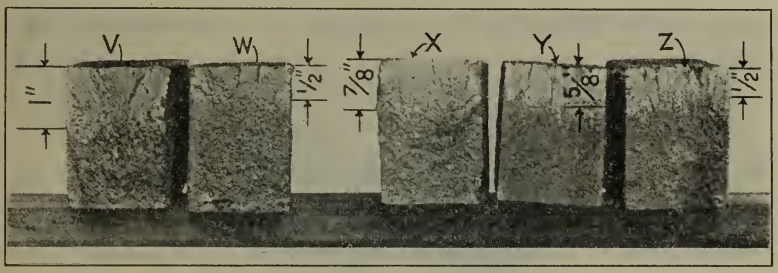

FIG. 7 SPECIMENS AFFECTED BY COOLING AND VARYING PERCENTAGES OF VANADIUM AND TITANIUM

iron, with bars 63 and 68 of Table 8. No comparison of the white iron in the round bars can be given, since there are no chilled round bars for this last set; but the contrast is so great in the square chilled bars 55 and 65, which show the white of the spurious metal, that it is safe to consider them 30 per cent weaker than the regular iron.

37 Some tests having shown vanadium and titanium beneficial in increasing the strength, it seems reasonable to suppose that all the other sets having them alloyed in the regular metal should show a similar tendency. It may be that in car-wheel metal there is a definite absorption of the alloys necessary to increase materially the strength of the gray and white. By the use of large round test bars for making the relative comparison in these irons, further experimenting with this grade of metal along practical lines should establish beyond any doubt the question of such a limit.

COMPARISON OF PARTLY CHILLED WITH WHOLLY CHILLED AND GRAY BODIES

38 In conducting this series of tests, bars were cast having only

one side chilled as companion bars to the all-chilled and gray bars, as seen by the second test, Tables 5 to 9. It will be a surprise to many to find that in all the tests, excepting the two of Set 14, the partly chilled bars are weaker than the all-chilled or white bars. A good view of these three companion bars is shown in Fig. 5, *K* being the partly chilled side.

39 The weakness of the partly chilled bars is due to internal strains and scattered amalgamation of the state of the broken carbon of the metals. Bars showing nearly every effect of rapid and slow cooling, and in no wise possessing the homogeneous blending of one character of grains, seen by the wholly white and gray, is well illustrated by *H*, *L*, Fig. 5.

40 All the partly chilled bars showed the chilled body interlac-

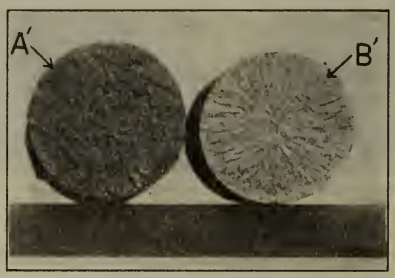

FIG. 8 SPECIMENS OF ALL CHILLED AND ALL SAND ROUND BARS CAST ON END

ing the mottled, and the latter blending into the gray, Figs. 5, 6 and 7. This is generally considered to be a stronger section than those where a distinct line marks the separation of the white and gray, and causes conditions such as can still further increase the weakness of partly chilled castings. Designers should duly consider this factor, so as either to have a strong backing of the mottled and the gray, or to have the chilled side of the casting arranged in compression if practical, when strains or concussions of its work is brought to bear upon it, a feature in keeping with the tests on treatment of Table 10.

COMPARISON OF STRENGTH IN ALL WHITE AND ALL GRAY IRONS

41 A feature of this paper worthy of consideration is the strength and deflection obtainable in strictly all-chilled or white iron. It is generally supposed that white iron is very much weaker than gray, and has very little if any deflection. By referring to Tests 7, 21, 30,

35, 44, 49, 54, 69, 82, 84 and 88, it will be seen that white iron can be obtained at least 75 per cent as strong as gray. White iron is the strongest with the crystals radiating from a center as at *M* and B,' Figs. 6 and 8. The round bar excels the square in this form of structure.

SPALLING WEAKNESS OF CHILLED OR WHITE IRON

42 The chief evil of white iron lies in its strength being erratic, and easily spalled. It is believed that founders could greatly increase and control the strength of various grades of white irons and make them much more reliable.[1]

43 Numerous experiments were conducted to test the spalling weakness of white and gray iron and it was found that white bodies do not possess much over one-third the strength to resist spalling blows that exists in the gray or mottled of the same iron. It shows the importance of designing that portion of the casting subject to such blows to contain so far as practicable gray or mottled iron.

HARDNESS TESTS OF ALL CHILLED, PARTLY CHILLED AND SAND-CAST TEST BARS

44 Table 12 gives Brinell and Scleroscope tests of three samples taken from each of the first three bars of Sets 12 to 19, a view of which is seen in Fig. 5. The Brinell depressions were produced by a $\frac{3}{8}$-in. ball loaded with 6000 lb. and the readings are the weight in kg. sustained by 1 mm. of area of the depression produced by the total load. This is the standard method of testing Brinell hardness. Both the Brinell and Scleroscope records are the averages of 4 to 6 tests on a sample.

45 The columns *H*, *I*, *J*, *K* and *L* give the tests of the surfaces indicated by the same letters shown in Fig. 5. Those who conduct these kinds of tests know that there is some variation of hardness over an area although it may not exceed 1 sq. in. The surface *H* had a variation of 3 to 7 points, *I*, 8 to 15 points, *J* about 6 points, while *L* had but 3 points, showing that a greater uniformity in hardness can be expected in all gray bodies than in mottled or chilled surfaces.

46 The table also shows that directly chilled faces, as *H*, are harder than those crystallizing over a sand surface caused by the heat-absorbing effect of a chiller, some distance from such points, as *I*. The excessive variation of the surface *I* is believed to be caused by the curved structure of the heat radiation lines, as they come to

[1] See footnote, p. 279.

the surface at an angle, differing from the straight lines shown on the sides at *H*.

47 The spurious iron is on an average harder than the regular iron. The alloys appear to have a hardening effect as compared with the regular iron, or that having only ferro-manganese in it. The irregularity in the effect of the alloy is no doubt due to the variations in the temperature of the metal filling the molds, and brings about variations in hardness similar to creating an irregularity in the chill, strength, deflection and contraction of like irons.

CREDIT FOR PROFESSIONAL COÖPERATION

48 Nearly all the bars tested for records in this paper were cast at the Cleveland plant of the National Car Wheel Company, and

TABLE 12 BRINELL AND SCLEROSCOPE HARDNESS TESTS OF SPECIMENS, FIG. 4

Set No.	Class	*H*		*I*		*J*		*K*		*L*	
		Bri.	Scl.	Bri.	Scl.	Bri.	Scl.	Bri.	Scl.	Bri.	Scl.
12	*R*	394	65	348	59	179	39	326	55	185	39
13	*V*	377	63	358	60	227	41	403	60	199	41
14	*R*	417	66	403	58	185	38	386	61	175	38
15	*T*	419	68	427	56	211	41	412	58	186	40
16	*S*	452	69	412	59	224	40	443	60	191	42
17	*R*	358	62	390	56	189	39	317	47	173	37
18	*TV*	382	64	375	61	193	40	400	54	183	40
19	*TV*	442	66	422	54	189	41	417	57	179	38

tested by Mr. H. E. Smith, engineer of tests, assisted by Mr. G. E. Doke, both of the Lake Shore & Michigan Southern Railroad, Collinwood, Ohio. Assistance was rendered in making the bars by Messrs. H. E. McClumpha, general manager, J. D. Cunningham, plant manager, and Charles K. Logue, foreman of the wheel foundry. Some of the tests shown herein were made by Prof. John H. Nelson of the Case School of Applied Science. The hardness tests were made by Mr. Robert R. Abbott, metallurgical engineer of the Peerless Motor Car Company, the analyses by Messrs. Crowell and Murray, chemists, both firms of Cleveland. The vanadium was furnished by Mr. George L. Norris, engineer of tests of the Vanadium Sales Company of America, and the titanium by Mr. Charles V. Slocum, general sales agent of the Titanium Alloy Manufacturing Company, both of Pittsburgh, Pa. All the test bars were made by the writer and tested under his supervision, who hereby desires to tender sincere thanks to all the above firms and gentlemen for having so kindly and ably assisted him in these researches.

A. E. Outerbridge, Jr.[1] In making comparative tests with powdered alloys in ladles of molten iron the writer always uses a cast-iron stirring rod, as he has found that a very appreciable increase in strength is imparted to cast iron by using a wrought-iron rod. He also has two sets of test molds, including long, thin fluidity strips 10 in. long by ¼ in. thick tapering to a feather edge, and uses two cleaned hand ladles, into which he pours the metal from a bull ladle. Both the treated and untreated samples are stirred with cast-iron rods and both poured at the same moment, so that all conditions are alike. Usually shrinkage test pieces are poured for the purpose of developing any tendency to produce "blow" holes at the junction of light and heavy sections. These methods date back to the time when he began and carried on long continued experiments on the effect of adding small quantities of powdered ferro-manganese to car wheel iron in the ladles, first published in the Journal of The Franklin Institute, March 1888.

C. B. Murray.[2] Ever since 1908, when Dr. Moldenke read his paper on the tests of vanadium in cast iron, there has been considerable discussion as to how vanadium and titanium do really affect such iron.

When vanadium first made its appearance, and began to be used in steel, it was thought, for a time, that its action was of the nature of a scavenger or a cleaner of the metal, and that the actual presence of vanadium in the finished steel was not necessary. This idea has, however, been cast one side, and now it is pretty generally believed that vanadium acts in two ways: first as a deoxidizer, the resultant vanadium oxide going into the slag; second, that a part of the vanadium should be left in the steel. Just how this acts is somewhat uncertain, but it seems to intensify the action of carbon and other metalloids in the steel. This same theory has been applied to titanium and seems to be still held.

Mr. West has been very painstaking, and seems to have attained his object of securing two specimens of as nearly as possible the same chemical and physical characteristics. The results of the tests, however, especially with regard to the modulus, do not give much information.

[1] Metallurgist, Wm. Sellers & Co., Inc., Phila., Pa.

[2] Crowell & Murray, Cleveland, O.

The determinations of small amounts of vanadium and titanium are somewhat difficult operations and require a great deal of care and attention. The chemical determinations of these materials were very carefully made in the laboratory of Crowell & Murray in Cleveland and the writer feels assured that the results are correct.

The writer has attempted to get some data from Tables 5-9, but apparently the most marked effects of the addition of these elements is in the chilled round samples. Just why this is so he is unable to say, but it is hardly a coincidence, since in all cases where the chilled round samples can be compared the iron containing either vanadium or titanium, or both, shows a marked increase in their modulus. Possibly the effect of all the carbon being in the combined state and the fact that Mr. West has apparently secured a more homogenous thoroughly chilled sample in the round piece than in the square pieces may have some bearing on this question.

His conclusions in Par. 37 would seem to indicate that he himself is not thoroughly satisfied with the results of the test, and that perhaps some conditions regarding the temperature of pouring, or the thoroughness of mixing, may have escaped his notice, and that these unsatisfactory results may have come from such conditions.

Henry Souther. The irregularity of the figures given by the author for the modulus of the iron suggests the possibility of internal strains in the cast-iron specimens. The question arises as to whether or not these strains were relieved by partial annealing. If not, it would seem that these great irregularities may be due to uncertain internal stress.

In reference to the effect of vanadium and titanium, it appears that these alloying elements were added in the ladle and that the metal cooled somewhat because of these additions. An indication of increase of strength was obtained. It is fair to assume that the hotter the metal the slower it will cool in the molds. It is a fact that the slower the cooling, the greater the separation of graphitic carbon and the weaker the specimen. After the addition of vanadium or titanium the metal is cooler, therefore chills more quickly with less separation of graphitic carbon and has greater strength. So far as the writer has noted, Mr. West

has not taken this into consideration. It is unfortunate that the iron with and without alloying elements was not poured at the same temperature.

One point the writer would like to see cleared up is the question of modulus. Various figures for the modulus appear in many of the tables. They vary widely one from the other and also from what is known of the modulus of the iron. It would be an addition to the paper could Mr. West supply the means of obtaining these figures; that is, it would be well to show the formula used and the methods taken to measure the modulus.

Bradley Stoughton. The writer is greatly interested in the effect of titanium on the strength, contraction and chilling of iron, because Mr. West's results confirm some hitherto unpublished tests of his own. Mr. West finds that titanium reduces the contraction and the tendency to chill, which is corroborated by the writer's observations. In an attempt to prevent chilling altogether, he added first 1.14 per cent of titanium, and then 2.29 per cent; since the latter percentage did not prevent chilling as effectively as the former, he concludes in Par. 31, "that iron could not be prevented from chilling beyond a certain limit by its use." The writer's investigations showed that 2 per cent of the usual titanium alloy was sufficient to convert even a dead white iron, in bars 1½ in. in diameter, into a mottled iron, cooled in sand, and that 3 per cent of the alloy was more effective than 2 per cent. In the quality of contraction, the writer's observations corroborate those of Mr. West; namely, that titanium reduces this property. As to the most important characteristic from the commercial standpoint, strength, Mr. West finds that titanium increases the strength of white cast iron by 27 and 32 per cent respectively. The writer's results show an increase in strength all the way from small amounts up to 43 per cent, depending upon the grade of iron, the amount of titanium added and the temperature of pouring. In this connection it may be interesting to note that the writer's investigations cover also the strength of gray cast irons of a great variety of grades, and that the increase produced by adding titanium varies up to 16 per cent.

Paul Kreuzpointner.[1] In determining the quality of a material of construction for a given purpose three factors have to

[1] Secy. Am. Fdymen's Assoc., Altoona, Pa.

be taken into consideration: (*a*) The nature of the material, (*b*) the nature of the forces tending to destroy the structure, and (*c*) the proper relation of the methods employed in testing to the properties of the material. Concerning the nature of the material, it is obvious that the mechanical means used in testing must conform to that nature. What is suitable for testing cement is not suitable for testing other materials.

It is not quite so simple to determine the best manner for testing a material in accordance with that in which it is subject to destruction when in service. The forces tending to destroy a structural material may be slow-acting or quick-acting; they may be transverse, bending, crushing or tensile forces or a combination of any complicated by variation of temperatures. Thus, chillable irons are chiefly used where they are subjected to crushing and bending stresses; hence the best manner of testing them is by the application of a crushing, bending and drop or pendulum test.

However much the qualities of structural materials of the same class may vary in accordance with the requirements for a given purpose, all metals have certain properties which are the same; that is to say all metals are elastic, ductile, strong in power of resistance. Each and all of these properties vary in degree but not in kind. This being so, it may be concluded that according to the third factor in testing materials, the methods must have uniformity in all such cases where the degree of the property of a metal determines its suitability for a given purpose. Thus, if ductility is the measure of quality of a metal for a given purpose, it is by no means a matter of indifference what size of test section is used nor whether the same section is used every time or not.

Erroneous results will surely be the result of testing whenever there is a departure from a standard uniform test section or a recognized standard width of support simply because it is convenient for the time being to vary the method.

As already said all metals are plastic though they vary in degree of plasticity, and to this is due the fact that all metals flow under pressure. But the degree of flow in a test piece under test is free or is influenced by the length of the test section or the width of supports or the distance of the grips. Hence in judging of the value of the results of a series of tests, as in

Mr. West's paper, for determining the quality of a metal for a given purpose, it should be ascertained at the same time whether the methods employed, speed of machine, test section, width of supports, etc., represent a recognized uniform standard.

THE AUTHOR. Comments have been made on the value of test bars. It is not so much that we have no information that can be compiled to guide us as it is the indifference and often inability of many to digest what we have. Many test bars are of little value for making comparisons or obtaining the relative physical qualities of cast iron unless properly made. Owing to the lack of proper methods for this work the writer considers the majority of tests to be questionable.

As an example, compare the methods illustrated in Figs. 1 and 2 where one bar provides for two or three tests, with casting one bar one day for one test and another bar for a second test another day. Under the latter condition there may be a variation in coke, iron and scrap, method of charging, blast pressure, temper of sands, methods and manner of holding, temperature of metals and speed of pouring, etc. The writer has given a great deal of study to the origination of methods best adapted for securing comparative results in the use of test bars. It is shown in the past methods advanced for making test bars as well as in those for obtaining the results given in the paper.

Mr. Souther comments on the evils of variation in pouring temperatures of metals for making comparative test bars. The writer labored to govern this factor, as far as conditions would permit. However, he believes that what irregularities did exist in this factor were not so great as to affect the results. This can be seen, for example, in the difference of strength of like bars having ferro-manganese in them, and those without it as recorded by the tests Nos. 55 to 59 when compared with Nos. 65 to 68.

In response to Mr. Souther's request that the formula be given and the method described for measuring the modulus, the following has been prepared by A. G. Smith, who coöperated with the author:

The question of modulus of rupture which Mr. Souther raises is quite important. We agree that it does not indicate the actual strength of the iron, and is, in fact, only a conventional figure for the reduction to a common basis of tests on similar but

slightly varying bars. For example, bars nominally two inches square may actually vary from 1.95 to 2.05 in. in one of both dimensions. If no allowance is made for this variation, the results may be very misleading. The modulus was calculated according to the following very familiar formulæ:

$$M = \frac{3}{2}\frac{Pl}{bd^2} \quad \text{for rectangular bars}$$

and

$$M = \frac{3}{2}\frac{Pl}{0.59D^3} \quad \text{for round bars}$$

where

M = modulus
P = breaking load
l = length of bar in inches (between supports)
b = breadth of bar in inches
d = depth of bar in inches
D = diameter of bar in inches

The actual measurements of the bars and the actual breaking loads are given with each table.

It is to be understood that none of the figures refer to modulus of elasticity; only to the so-called modulus of rupture.

No. 1352

STRENGTH OF STEEL TUBES, PIPES AND CYLINDERS UNDER INTERNAL FLUID PRESSURE

By Reid T. Stewart, Pittsburgh, Pa.
Member of the Society

PART I A COMPARISON OF THE THEORETICAL FORMULAE

In order to arrive at some definite conclusion as to what formula or formulae should be used for calculating the strength of tubes, pipes and cylinders subjected to internal fluid pressure, the different published formulae have been investigated and compared. These are five in number, namely, the common formula, and those by Barlow, Lamé, Clavarino, and Birnie.

2 These formulae have been put into the simplest form for application to tubes, pipes and cylinders, and are reduced to a common notation for the sake of making an easy comparison. The notation used is as follows:

D_1 = outside diameter in inches.
D_2 = inside diameter in inches.
t = thickness of wall in inches.
p = internal gage pressure, or difference between internal and external fluid pressures, in lb. per sq. in.
f = fiber stress in the wall in lb. per sq. in.

3 The formulae here given are for the usual conditions of practise, namely, where the external pressure is atmospheric and the internal pressure is expressed as gage pressure. They are also applicable to cases where the external pressure is not excessive by taking p as the difference between the internal and external pressures.

4 In all that follows it is assumed that the length of the tube or pipe relative to its diameter is sufficiently great to eliminate the influence of end supports tending to prevent rupture.

5 *Nature of Stress in Tube Wall.* An interanl fluid pressure may give rise (*a*) to a circumferential stress within the wall of a tube or pipe, or (*b*) to both a circumferential and a longitudinal stress acting

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jointly. In either case the tube wall is under radial compressive stress as indicated by the arrows, Figs. 1 and 2.

6 Fig. 1 illustrates a tube with frictionless plungers fitted into its ends, the plungers being kept in place by the external forces, PP, which exactly balance the internal fluid pressure tending to force them outward. In this case the tube wall is subjected only to the internal forces shown as acting at right angles to its inner surface. It is obvious that these forces can give rise to radial and circumferential stresses only in the tube wall. The value of the circumferential stress, f_t, in lb. per sq. in. is

$$f_t = p\frac{D_2}{D_1 - D_2} = \frac{pD_2}{2t} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

7 Fig. 2 illustrates the ordinary case of a tube or pipe with both ends closed. In this case the tube wall, as in Fig. 1, is subjected to the circumferential stress, f_t, along with the radial stress, and at the same time is subjected to the longitudinal stress, f_l. The longitudinal

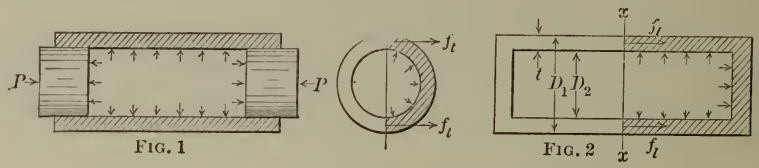


Fig. 1

Fig. 2

Fig. 1 Tube with Frictionless Plungers

Fig. 2 Tube with Both Ends closed

stress is caused by the internal fluid pressure tending to force the attached heads outward and expressed in lb. per sq. in. is

$$f_l = p\frac{D_2^2}{4(D_2 + t)t} \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

When the thickness of wall t, is relatively small with respect to the diameter, the longitudinal stress becomes approximately

$$f_l = \frac{pD_2}{4t} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

or one-half the corresponding circumferential stress.

8 *Common Formula.* This is the formula generally found in books on mechanics. It is based on the condition that the tube wall is subjected to circumferential stress only, Fig. 1, and assumes (*a*) that the material of the tube wall is devoid of elasticity, and (*b*) that the stress is the same on all the circumferential fibers from the innermost to the outermost. These assumptions are only approxi-

mately true for tubes of comparatively thin walls, and are greatly in error for tubes having very thick walls.

9 Using the notation as given above, the formula is

$$\frac{p}{f}=2\frac{t}{D_2};\quad p=2f\frac{t}{D_2};\quad t=\tfrac{1}{2}D_2\frac{p}{f};\quad f=\tfrac{1}{2}D_2\frac{p}{t}\qquad[4]$$

10 Referring to the charts, Figs. 3 and 4, it will be seen that the common formula gives quite close results for comparatively thin walls when used for the conditions shown in Fig. 1, for which Birnie's formula is theoretically correct. The error increases as the thickness of wall becomes relatively greater, reaching 10 per cent for a thickness ratio $\frac{t}{D_1}$, of about 0.05. For thick walls the error is great; for example, when $\frac{t}{D_1}=0.25$ the value of $\frac{p}{f}$ is about 100 per cent in error. It should be observed when applying the common formula to this case that the error is always on the side of danger.

11 For the conditions shown in Fig. 2, that is, when the tube is subjected to the stresses due to an internal fluid pressure acting jointly on the tube wall and its closed ends, for which Clavarino's formula is theoretically correct, the curves show for a thickness ratio, $\frac{t}{D_1}$, less than 0.07 that the common formula errs on the side of safety, the greatest error being about 12 per cent; while for thickness ratios greater than 0.07 the error is on the side of danger, reaching 10 per cent for a thickness ratio of 0.1 and about 100 per cent for a ratio of 0.25.

12 *Barlow's Formula.* This formula assumes (*a*) that because of the elasticity of the material, the different circumferential fibers will have their diameters increased in such a manner as to keep the area of cross-section constant; and (*b*) that the length of the tube is unaltered by the internal fluid pressure. As neither of these assumptions is theoretically correct, this formula can give only approximately correct results. Using the notation given above, this formula is

$$\frac{p}{f}=2\frac{t}{D_1};\quad p=2f\frac{t}{D_1};\quad t=\tfrac{1}{2}D_1\frac{p}{f};\quad f=\tfrac{1}{2}D_1\frac{p}{t}\qquad[5]$$

13 It should be observed that while Barlow's formula is similar in form to the common formula, it gives results that are quite different when applied to tubes, pipes, and cylinders having walls of considerable thickness. This is due to the fact that Barlow's formula

is expressed in terms of the outside diameter, D_1, whereas the common formula is expressed in terms of the inside diameter, D_2.

14 Referring to Figs. 3 and 4, it will be seen that Barlow's for-

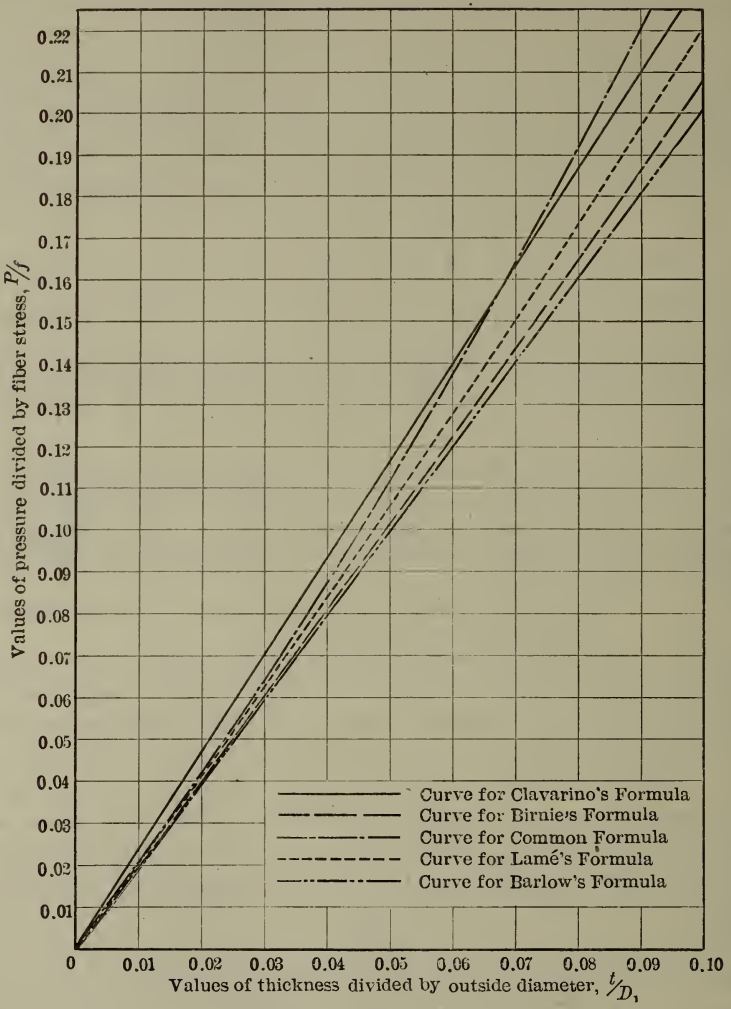


FIG. 3 COMPARISON OF INTERNAL FLUID PRESSURE FORMULAE FOR TUBES, PIPES AND CYLINDERS

mula gives quite close results when used for the condition shown in Fig. 1, for which Birnie's formula is theoretically correct. The curves show for the entire practical range of thickness ratios that

the error in values of $\frac{p}{f}$, for this case, does not exceed 3 per cent, the error throughout the whole practical range being on the side of

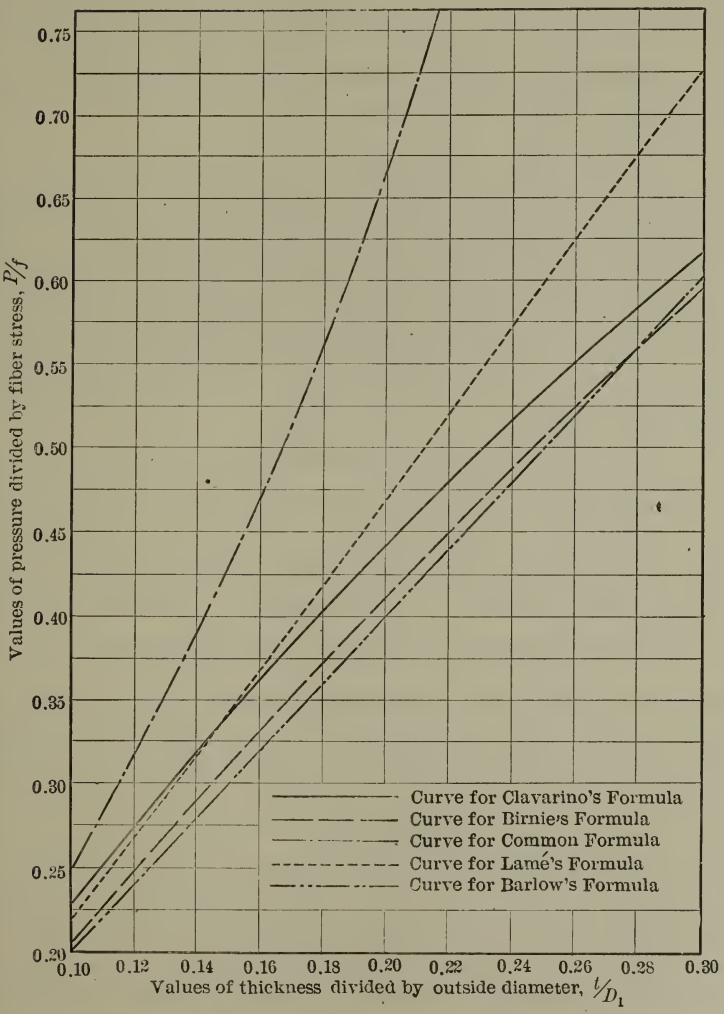


FIG. 4 COMPARISON OF INTERNAL FLUID PRESSURE FORMULAE FOR TUBES, PIPES AND CYLINDERS

safety. This then is the best of the simple theoretical formulae for application to the cases illustrated in Fig. 1.

15 For the conditions shown in Fig. 2, namely when the tube is subjected to the stresses due to an internal fluid pressure acting jointly on the tube wall and its closed ends, for which Clavarino's formula is theoretically correct, the curves show that Barlow's formula gives values of $\frac{p}{f}$ whose errors range from 15 per cent for tubes, pipes and cylinders having thin walls, to 10 per cent for those having thick walls, the error being on the side of safety for all practical thickness ratios.

16 *Lamé's Formula.* This formula is meant to apply to the conditions shown in Fig. 2. Each material particle of the tube wall is supposed to be subjected to the radial compression, the circumferential and longitudinal tensions due to an internal fluid pressure acting jointly on the tube wall and its closed ends; and the material of the tube wall is supposed to be elastic under these actions. Lamé's formula, however, ignores the coefficient of lateral contraction, known as Poisson's Ratio, and consequently is not theoretically correct.

17 Using the notation as given above this formula is

$$\frac{p}{f}=\frac{D_1^2-D_2^2}{D_1^2+D_2^2};\quad p=\frac{D_1^2-D_2^2}{D_1^2+D_2^2}f;\quad D_2=D_1\sqrt{\frac{f-p}{f+p}};\quad D_1=D_2\sqrt{\frac{f+p}{f-p}}\ldots[6]$$

18 Referring to Figs. 3 and 4, it will be seen that Lamé's formula, which is meant to apply to the conditions for which Clavarino's formula is theoretically correct, gives for thickness ratios, $\frac{t}{D_1}$, less than 0.15, an error on the side of safety, the error having a maximum value of about 14 per cent when $\frac{t}{D_1}$ equals 0.01. For thickness ratios greater than 0.15 the error is on the side of danger, reaching 10 per cent for a ratio of about 0.23.

19 *Clavarino's Formula.* In this formula, as in Lamé's formula, each material particle of the tube wall is supposed to be subjected to the radial compression and the circumferential and longitudinal tensions due to an internal fluid pressure acting jointly on the tube wall and its closed ends; and the material is supposed to be elastic under these actions. Unlike Lamé's formula, however, this formula expresses the true stresses in the tube wall as based upon the coefficient of lateral contraction, known as Poisson's ratio, and is consequently theoretically correct for the conditions shown in Fig. 2,

providing the stress on the most strained fiber does not exceed the elastic limit of the material.

20 Using the notation given above and assuming the value of the coefficient of lateral contraction for tube steel to be 0.3, this formula is

$$\frac{p}{f}=\frac{10(D_1^2-D_2^2)}{13D_1^2+4D_2^2};\quad p=\frac{10(D_1^2-D_2^2)}{13D_1^2+4D_2^2}f;\quad D_1=D_2\sqrt{\frac{10f+4p}{10f-13p}};$$

$$D_2=D_1\sqrt{\frac{10f-13p}{10f+4p}}\qquad\qquad[7]$$

21 This theoretically correct formula for the conditions shown in Fig. 2 has the disadvantage that it is difficult to apply directly in making calculations. In order to remove this difficulty Table 1 has been prepared, by means of which any desired calculation can be as readily made by Clavarino's formula as by any of the simpler formulae. The entries of this table are the values in Clavarino's formula of the factor

$$\frac{10(D_1^2-D_2^2)}{13D_1^2+4D_2^2}=K$$

22 It will be observed that these factors are tabulated for thickness ratios, $\frac{t}{D_1}$, from 0.01 to 0.3, advancing by thousandths. Thus for a wall thickness, t, of 0.25 in. and an outside diameter, D_1, of 10 in., the thickness ratio, $\frac{t}{D_1}$, would be 0.25 divided by 10, or 0.025. The required factor corresponding to this thickness ratio is 0.0587 and is found in the column headed 0.005 opposite 0.02 in column 1. Similarly for an outside diameter of 4 in. and a wall thickness of 0.5 in., the thickness ratio would be 0.125 and the corresponding internal pressure factor is 0.2869.

23 If we designate the value of any tabular factor by K, then it is obvious that Clavarino's formula may be written

$$\frac{p}{f}=K;\quad p=Kf;\quad f=\frac{p}{K}\qquad\qquad[8]$$

24 Table 1 is well adapted to the ready solution of problems involving the strength and safety of a tube, pipe or cylinder which is subjected to the stresses due to an internal fluid pressure acting jointly on its wall and closed ends, as illustrated in Fig. 2.

Problem 1. Required the safe working fluid pressure, p (Fig. 2), when the outside diameter, $D_1=4$ in.; thickness of wall, $t=0.5$ in.; and the working fiber

stress of the steel, $f = 10{,}000$ lb. Solution: (*a*) the thickness ratio, $\frac{t}{D_1} = 0.125$; (*b*) the corresponding tabular factor, K, is found from Table 1 to be 0.2869; and (*c*) the required safe working fluid pressure, $p = Kf$, equation [8], or $0.2869 \times 10{,}000$, or 2869 lb. per sq. in.

TABLE 1 INTERNAL FLUID PRESSURE FACTORS, K, FOR CONDITIONS SHOWN IN FIG. 2

CALCULATED BY CLAVARINO'S FORMULA, ASSUMING FOR STEEL A COEFFICIENT OF LATERAL CONTRACTION (POISSON'S RATIO) = 0.3.

RULE: DIVIDE THICKNESS OF TUBE OR PIPE BY ITS OUTSIDE DIAMETER, BOTH BEING EXPRESSED IN INCHES, THEN MULTIPLY THE TABULAR VALUE CORRESPONDING TO THIS QUOTIENT BY THE WORKING FIBER STRESS IN LB. PER SQ. IN. THE RESULT WILL BE THE SAFE INTERNAL PRESSURE IN LB. PER SQ. IN.

$\frac{t}{D_1}$	0.000	0.001	0.002	0.003	0.004	0.005	0.006	0.007	0.008	0.009
0.01	0.0235	0.0259	0.0282	0.0306	0.0329	0.0352	0.0376	0.0399	0.0423	0.0446
0.02	0.0470	0.0493	0.0517	0.0540	0.0564	0.0587	0.0610	0.0634	0.0657	0.0681
0.03	0.0704	0.0727	0.0751	0.0774	0.0797	0.0821	0.0844	0.0867	0.0891	0.0914
0.04	0.0937	0.0961	0.0984	0.1007	0.1031	0.1054	0.1077	0.1100	0.1122	0.1147
0.05	0.1170	0.1193	0.1216	0.1239	0.1263	0.1286	0.1309	0.1332	0.1355	0.1378
0.06	0.1401	0.1424	0.1448	0.1471	0.1494	0.1517	0.1540	0.1563	0.1586	0.1609
0.07	0.1632	0.1655	0.1678	0.1700	0.1723	0.1746	0.1769	0.1792	0.1815	0.1838
0.08	0.1861	0.1883	0.1906	0.1929	0.1952	0.1974	0.1997	0.2020	0.2043	0.2065
0.09	0.2088	0.2111	0.2133	0.2156	0.2178	0.2201	0.2223	0.2246	0.2269	0.2291
0.10	0.2314	0.2336	0.2358	0.2381	0.2403	0.2425	0.2448	0.2470	0.2493	0.2515
0.11	0.2537	0.2559	0.2582	0.2604	0.2626	0.2648	0.2670	0.2692	0.2715	0.2737
0.12	0.2759	0.2781	0.2803	0.2825	0.2847	0.2869	0.2890	0.2912	0.2934	0.2956
0.13	0.2978	0.3000	0.3022	0.3043	0.3065	0.3087	0.3108	0.3130	0.3152	0.3173
0.14	0.3195	0.3216	0.3238	0.3259	0.3281	0.3302	0.3323	0.3345	0.3366	0.3388
0.15	0.3409	0.3430	0.3451	0.3472	0.3494	0.3515	0.3536	0.3557	0.3578	0.3599
0.16	0.3620	0.3641	0.3662	0.3683	0.3704	0.3724	0.3745	0.3766	0.3787	0.3808
0.17	0.3828	0.3849	0.3869	0.3890	0.3910	0.3931	0.3951	0.3972	0.3992	0.4013
0.18	0.4033	0.4053	0.4073	0.4094	0.4114	0.4134	0.4154	0.4174	0.4194	0.4214
0.19	0.4234	0.4254	0.4274	0.4294	0.4314	0.4333	0.4353	0.4373	0.4393	0.4412
0.20	0.4432	0.4452	0.4471	0.4490	0.4510	0.4529	0.4548	0.4568	0.4587	0.4606
0.21	0.4626	0.4645	0.4664	0.4683	0.4702	0.4721	0.4740	0.4758	0.4777	0.4796
0.22	0.4815	0.4834	0.4852	0.4871	0.4889	0.4908	0.4926	0.4945	0.4964	0.4982
0.23	0.5001	0.5019	0.5037	0.5055	0.5073	0.5091	0.5109	0.5127	0.5145	0.5163
0.24	0.5181	0.5199	0.5216	0.5234	0.5252	0.5269	0.5287	0.5304	0.5322	0.5340
0.25	0.5357	0.5374	0.5391	0.5408	0.5426	0.5443	0.5460	0.5477	0.5494	0.5511
0.26	0.5528	0.5545	0.5561	0.5578	0.5594	0.5611	0.5628	0.5644	0.5661	0.5677
0.27	0.5694	0.5710	0.5726	0.5742	0.5758	0.5774	0.5790	0.5806	0.5822	0.5838
0.28	0.5854	0.5870	0.5885	0.5901	0.5916	0.5932	0.5947	0.5963	0.5978	0.5994
0.29	0.6009	0.6024	0.6039	0.6054	0.6069	0.6084	0.6099	0.6114	0.6129	0.6143
0.30	0.6158	0.6173	0.6187	0.6201	0.6216	0.6230	0.6244	0.6259	0.6273	0.6287

Problem 2. Required the fiber stress, f, in the wall of a cylinder, Fig. 2, when the outside diameter, $D_1 = 5.5$ in., the thickness of wall, $t = 0.25$ in.; and the working fluid pressure, $p = 1500$ lb. per sq. in. Solution: (*a*) the thickness ratio, $\frac{t}{D_1} = 0.045$; (*b*) the corresponding tabular factor, K, is found from Table 1 to be .1054; and (*c*) the required fiber stress, $f = \frac{p}{K}$, equation [8], or $1500 \div 0.1054$ = 14,200 lb. per sq. in.

Problem 3. Required the thickness of wall, t (Fig. 2), when the outside diameter, $D_1 = 8$ in.; the working fiber stress of the steel, $f = 15{,}000$ lb. per sq. in.; and the working fluid pressure, $p = 2000$ lb. per sq. in. Solution: (*a*) the factor, $K = \frac{p}{f}$, equation [8], or $2000 \div 15{,}000$, or 0.133; (*b*) the value of the thickness ratio, $\frac{t}{D_1}$, corresponding to this value of K is found from Table 1 to be 0.057;

TABLE 2 INTERNAL FLUID PRESSURE FACTORS, K, FOR CONDITIONS SHOWN IN FIG. 1

CALCULATED BY BIRNIE'S FORMULA, ASSUMING FOR STEEL A COEFFICIENT OF LATERAL CONTRACTION (POISSON'S RATIO) = 0.3.

RULE: DIVIDE THICKNESS OF TUBE OR PIPE BY ITS OUTSIDE DIAMETER, BOTH BEING EXPRESSED IN INCHES, THEN MULTIPLY THE TABULAR VALUE CORRESPONDING TO THIS QUOTIENT BY THE WORKING FIBER STRESS IN LB. PER SQ. IN. THE RESULT WILL BE THE SAFE INTERNAL PRESSURE IN LB. PER SQ. IN.

$\frac{t}{D_1}$	0.000	0.001	0.002	0.003	0.004	0.005	0.006	0.007	0.008	0.009
0.01	0.0201	0.0221	0.0241	0.0261	0.0282	0.0302	0.0322	0.0342	0.0363	0.0383
0.02	0.0403	0.0423	0.0444	0.0464	0.0485	0.0505	0.0525	0.0546	0.0566	0.0586
0.03	0.0607	0.0627	0.0648	0.0668	0.0689	0.0709	0.0730	0.0750	0.0771	0.0791
0.04	0.0812	0.0832	0.0853	0.0873	0.0894	0.0915	0.0935	0.0956	0.0976	0.0997
0.05	0.1018	0.1038	0.1059	0.1080	0.1100	0.1121	0.1142	0.1163	0.1183	0.1204
0.06	0.1225	0.1245	0.1266	0.1287	0.1308	0.1329	0.1349	0.1370	0.1391	0.1412
0.07	0.1433	0.1453	0.1474	0.1495	0.1516	0.1537	0.1558	0.1579	0.1599	0.1620
0.08	0.1641	0.1662	0.1683	0.1704	0.1725	0.1746	0.1767	0.1787	0.1808	0.1829
0.09	0.1850	0.1871	0.1892	0.1913	0.1934	0.1955	0.1976	0.1997	0.2018	0.2039
0.10	0.2059	0.2080	0.2101	0.2122	0.2143	0.2164	0.2185	0.2206	0.2227	0.2248
0.11	0.2269	0.2290	0.2311	0.2332	0.2353	0.2374	0.2395	0.2416	0.2437	0.2457
0.12	0.2478	0.2499	0.2520	0.2541	0.2562	0.2583	0.2604	0.2625	0.2646	0.2667
0.13	0.2688	0.2708	0.2729	0.2750	0.2771	0.2792	0.2813	0.2834	0.2854	0.2875
0.14	0.2896	0.2917	0.2938	0.2959	0.2979	0.3000	0.3021	0.3042	0.3062	0.3083
0.15	0.3104	0.3125	0.3145	0.3166	0.3187	0.3208	0.3228	0.3249	0.3270	0.3290
0.16	0.3311	0.3332	0.3352	0.3373	0.3393	0.3414	0.3434	0.3455	0.3476	0.3496
0.17	0.3517	0.3537	0.3558	0.3578	0.3598	0.3619	0.3639	0.3660	0.3680	0.3700
0.18	0.3721	0.3741	0.3761	0.3782	0.3802	0.3822	0.3842	0.3863	0.3883	0.3903
0.19	0.3923	0.3943	0.3963	0.3983	0.4003	0.4024	0.4044	0.4064	0.4084	0.4104
0.20	0.4124	0.4144	0.4163	0.4183	0.4203	0.4223	0.4243	0.4262	0.4282	0.4302
0.21	0.4322	0.4341	0.4361	0.4380	0.4400	0.4419	0.4439	0.4459	0.4478	0.4498
0.22	0.4517	0.4536	0.4556	0.4575	0.4594	0.4613	0.4633	0.4652	0.4671	0.4690
0.23	0.4710	0.4729	0.4748	0.4767	0.4785	0.4804	0.4823	0.4842	0.4861	0.4880
0.24	0.4899	0.4918	0.4936	0.4955	0.4973	0.4992	0.5010	0.5029	0.5048	0.5066
0.25	0.5085	0.5103	0.5121	0.5139	0.5157	0.5176	0.5194	0.5212	0.5230	0.5248
0.26	0.5266	0.5284	0.5302	0.5320	0.5338	0.5355	0.5373	0.5391	0.5409	0.5427
0.27	0.5444	0.5462	0.5479	0.5496	0.5514	0.5531	0.5548	0.5566	0.5583	0.5600
0.28	0.5617	0.5634	0.5651	0.5668	0.5685	0.5702	0.5718	0.5735	0.5752	0.5769
0.29	0.5786	0.5802	0.5818	0.5835	0.5851	0.5867	0.5884	0.5900	0.5916	0.5933
0.30	0.5949	0.5965	0.5981	0.5996	0.6012	0.6028	0.6044	0.6059	0.6075	0.6091

and (*c*) the required thickness will result from multiplying this thickness ratio, $\frac{t}{D_1}$, by the outside diameter, D_1, or $0.057 \times 8 = 0.456$ in.

Note—When the inside diameter, D_2, the internal pressure, p, and the working fiber stress, f, are given and it is required to find the thickness of wall, t, proceed

by finding first the value of the outside diameter, D_1, by means of equation [7]; after which the required thickness may be had by taking one-half the difference of the outside and inside diameters or

$$t = \frac{D_1 - D_2}{2} \quad [9]$$

25 *Birnie's Formula.* This formula is based upon the conditions illustrated in Fig. 1. In its derivation precisely the same assumptions are made as for Clavarino's formula, with the single exception that the longitudinal stress, f_1, due to the internal fluid pressure acting upon attached heads, is assumed not to exist. Birnie's formula consequently is theoretically correct for tubes, pipes and cylinders that are subjected to an internal fluid pressure in such manner as not to give rise to longitudinal stress in the wall; provided the stress on the most strained fiber does not exceed the elastic limit of the material.

26 Using the same notation as before and assuming the value of the coefficient of lateral contraction for steel to be 0.3, this formula is

$$\frac{p}{f} = \frac{10(D_1^2 - D_2^2)}{13D_1^2 + 7D_2^2}; \quad p = \frac{10(D_1^2 - D_2^2)}{13D_1^2 + 7D_2^2} f; \quad D_1 = D_2 \sqrt{\frac{10f + 7p}{10f - 13p}};$$

$$D_2 = D_1 \sqrt{\frac{10f - 13p}{10f + 7p}} \quad [10]$$

27 Birnie's formula, like Clavarino's formula, has the disadvantage of being difficult to apply directly in making calculations. In order to remove this difficulty Table 2 has been prepared, the entries being the values in Birnie's formula of the factor

$$\frac{10(D_1^2 - D_2^2)}{13D_1^2 + 7D_2^2} = K$$

28 This table is used in a manner precisely similar to the table of factors for Clavarino's formula.

PART II RESULTS OF TESTS ON COMMERCIAL TUBES AND PIPES, AND THE APPLICABILITY OF THE THEORETICAL FORMULAE

29 In Part I there appears a full statement of the basis of each of the five theoretical formulae for the strength of tubes, pipes and cylinders, when subjected to internal fluid pressures, together with a comparison of results obtained by their use. One or the other of these formulae, taken apparently at random, has often been used without sufficient understanding of their application to practical conditions. It is the purpose of what follows to illustrate the proper application of these formulae, making use of the results of hydro-

static tests recently made on commercial pipes at one of the mills of the National Tube Company.

30 *Yield Point Tests on Commercial Pipe.* Tests were made under Clavarino's condition, Fig. 2, on 195 specimens of 10-in. and 279 specimens of 12-in. lap-welded steel pipes, all of which were made up into cylinders with heads welded to the pipe. The hydrostatic pressure was raised until the yield point of the material was reached. The unit stresses on the most strained fibers were then calculated by means of Clavarino's formula, the pipes having been measured by micrometer before welding in the head, to determine the least thickness of wall.

31 The average results of the yield points of the most strained fibers of the material constituting these pipes, when compared with the average yield point of tensile test specimens cut from about 400 similar pipes, may be summarized as follows:

Outside diameter of pipe, in	10.00	12.00
Least thickness of wall, in	0.172	0.164
Hydrostatic pressure at yield point, lb. per sq. in	1435	1195
Yield point by Clavarino's formula, lb. per sq. in	35,600	37,100
Yield point average of tensile tests, lb. per sq. in	37,000	37,000
Apparent error in yield point by Clavarino's formula, per cent	−3.8	+0.3

32 This summary of the average results of 474 tests is a very satisfactory confirmation of the accuracy of Clavarino's formula when applied to commercial steel pipes for the conditions under which the formula theoretically applies.

33 Other tests show that when the heads are attached to the pipe, as in Fig. 2, it lengthens upon application of an internal fluid pressure and that when the heads are held independently, as in Fig. 1, it shortens in accord respectively with the assumptions which constitute the basis of Clavarino's and Birnie's formulae regarding change of length under internal fluid pressure.

34 *Applicability of Clavarino's and Birnie's Formulae.* The above summary of results of tests on pipes shows that Clavarino's formula is applicable to commercial wrought-steel pipe for the conditions shown in Fig. 2, when the yield point of the most strained fiber is not exceeded and the least thickness of wall is accurately known.

35 Tests made at the Watertown Arsenal in 1892, 1893, 1894, 1897 and 1902 on sections of steel guns, show that Birnie's formula for the condition shown in Fig. 1, when applied up to the elastic limit of the most strained fiber, gives results which agree with the results of direct tests that are within the ordinary range of experi-

mental error. These Watertown Arsenal tests were all made on tubes the material and dimensions of which were uniform to a degree obtainable only by boring and turning from forgings of the choicest portion of selected ingots.

TABLE 3-a BURSTING TESTS OF COMMERCIAL TUBES AND PIPES

Size	No. of Pieces Burst	Nominal External Diameter, In.	Average Thickness of Walls, In.	Bursting Pressure Pounds per Square Inch			Head Condition*	No. Burst by Failure of Material not at Weld	Average Fiber Stress by Barlow's Formula	Class of Material
				Minimum	Maximum	Average				
Steel, Butt-Welded										
⅛	10	0.405	0.066	11840	17320	14266	*C*	1	44011	Standard pipe
¼	10	0.540	0.085	8830	14680	12206	*C*	1	38645	Standard pipe
⅜	10	0.675	0.088	5850	13030	10330	*C*	1	39272	Standard pipe
½	10	0.840	0.101	11380	16310	14038	*C*	0	58163	Standard pipe
¾	10	1.050	0.109	7150	9150	8020	*C*	0	38657	Standard pipe
1	10	1.315	0.131	4500	8800	6990	*C*	0	35085	Standard pipe
1¼	10	1.660	1.139	4400	7300	5808	*C*	0	34603	Standard pipe
1¼	15	1.660	0.140	5500	11900	7700	*C*	1	45215	Redrawn
1½	10	1.900	0.143	3000	6100	4960	*C*	0	33031	Standard pipe
2	11	2.375	0.149	3830	6060	4951	*C*	0	40485	Standard pipe
2½	10	2.875	0.198	4310	5740	5134	*C*	0	37351	Standard pipe
3	10	3.500	0.204	4650	6370	5398	*C*	0	46234	Standard pipe
1¼	10	1.660	0.180	7910	14280	10514	*C*	0	48922	Extra strong
2	10	2.375	0.213	7250	8940	8238	*C*	0	45935	Extra strong
2	10	2.375	0.220	6160	8920	7661	*C*	0	41347	Extra strong
	10	2.375	0.445	8500	18314	14992	*C*	0	40023	X X strong
							General average		41686	
Steel Lap-Welded										
2	10	2.375	0.155	4890	7940	6645	*C*	1	50962	Standard pipe
2	10	2.375	0.182	4860	10060	7361	*C*	0	47889	Standard pipe
3	10	3.500	0.210	3830	8200	6368	*C*	7	53560	Standard pipe
4	10	4.500	0.232	4810	5680	5249	*C*	1	51462	Standard pipe
5	10	5.563	0.258	3410	5260	4538	*C*	1	48882	Standard pipe
6	5	6.625	0.275	2450	5210	4088	*C*	0	49286	Standard pipe
6	5	6.625	0.275	3170	4760	3666	*B*	0	44106	Standard pipe
10	5	10.750	0.349	3560	4730	4290	*C*	1	66080	Standard pipe
10	5	10.750	0.347	2770	3940	3396	*B*	2	52692	Standard pipe
2	10	2 375	0.218	2500	9870	7909	*C*	0	43254	Extra strong
2	10	2.000	0.108	5100	6560	6062	*C*	7	55607	Boiler tubes
3	10	3.000	0.112	3220	4860	3967	*C*	1	52957	Boiler tubes
4	5	4.000	0.135	3640	4070	3840	*C*	2	56978	Boiler tubes
4	5	4.000	0.136	3720	4040	3914	*B*	1	57440	Boiler tubes
							General average		52225	

* *C* = Clavarino conditions, Fig. 2. *B* = Birnie conditions, Fig. 1.

36 It is apparent that any variation below the nominal or average value in strength of material, thickness of wall and efficiency of joint in welded pipe, or above the nominal in diameter, will give results which err on the side of danger when making use of either Clavarino's

or Birnie's formulae. These formulae then should be restricted in their use to certain classes of seamless tubes and cylinders and to critical examinations of ordinary tubes, pipes and cylinders, when exact results are desired and sufficiently accurate data are available.

37 For all ordinary calculations of strength of commercial tubes, pipes and cylinders, Barlow's simple approximate formula, equation [5], is preferable.

38 *Bursting Tests of Commercial Tubes and Pipes.* Tables 3-a, 3-b

TABLE 3-b BURSTING TESTS OF COMMERCIAL TUBES AND PIPES

Size	No. of Pieces Burst	Nominal External Diameter, In.	Average Thickness of Walls, In.	Bursting Pressure Pounds per Square Inch			Head Condition*	No. Burst by Failure of Material not at Weld	Average Fiber Stress by Barlow's Formula	Class of Material
				Minimum	Maximum	Average				
Steel, Seamless										
2	10	2.000	0.098	5420	6590	6052	*C*	10	61530	Boiler tubes
3	10	3.000	0.112	3940	4730	4272	*C*	10	57075	Boiler tubes
4	6	4.000	0.134	4160	4440	4318	*C*	6	64450	Boiler tubes
4	4	4.000	0.134	4250	4440	4328	*B*	4	64488	Boiler tubes
							General average		61886	
Iron, Butt-Welded										
1¼	10	1.660	0.136	2880	6290	5283	*C*	3	32126	Standard pipe
1¼	10	1.660	0.136	3640	5680	4891	*C*	1	29817	Standard pipe
2	10	2.375	0.156	2930	4250	3687	*C*	2	28051	Standard pipe
1¼	10	1.660	0.188	2770	7330	5895	*C*	1	26678	Extra strong
							General average		29168	
Iron, Lap-Welded										
2	10	2.375	0.152	2400	3940	3213	*C*	1	25122	Standard pipe
2	10	2.375	0.207	5530	7120	6349	*C*	8	36461	Extra strong
							General average		30792	

* *C* = Clavarino conditions, Fig. 2. *B* = Birnie conditions, Fig. 1

and 4 show the average results of several hundred tests of commercial tubes and pipes, all of which were burst by hydrostatic pressure at one of the mills of the National Tube Company. Of these 95 per cent was made by this company, while 86 per cent of the wrought-iron pipe tested was obtained by purchase in the open market.

39 The average ultimate tensile strength of pipe steel is 57,000 lb. per sq. in., whether taken in the direction of rolling or transversely thereto, while that of the seamless steel tested is 60,000 lb. per sq. in. No tensile tests were made of the material of the wrought-iron pipes.

40 An examination of these tables will lead to the following general conclusions:

a In commercial welded pipe the variations in thickness of

TABLE 4 STRENGTH OF WELDS OF COMMERCIAL TUBES AND PIPES SELECTED FROM PRECEDING TABLE OF BURSTING TESTS

Size	No. Pieces Bursting in Weld[1]	Average Fiber Stress by Barlow's Formula	Class of Material
Steel, Butt-Welded			
⅛	9	43938	Standard pipe
¼	9	37777	Standard pipe
⅜	9	38954	Standard pipe
½	10	58163	Standard pipe
¾	10	38657	Standard pipe
1	10	35085	Standard pipe
1¼	10	34603	Standard pipe
1¼	14	45643	Redrawn
1½	10	33031	Standard pipe
2	11	40485	Standard pipe
2½	10	37351	Standard pipe
3	10	46234	Standard pipe
1¼	10	48922	Extra strong
2	10	45935	Extra strong
2	10	41347	Extra Strong
2	10	40023	X X Strong
General average		41634	
Steel, Lap-Welded			
2	9	50052	Standard pipe
2	10	47889	Standard pipe
3	3	54510	Standard pipe
4	9	51019	Standard pipe
5	9	48852	Standard pipe
6	10	47026	Standard pipe
10	7	59537	Standard pipe
2	10	43254	Extra Strong
2	3	56933	Boiler tubes
3	9	51980	Boiler tubes
4	7	57521	Boiler tubes
General average		51688	
Iron, Butt-Welded			
1¼	7	31136	Standard pipe
1¼	9	30680	Standard pipe
2	8	27323	Standard pipe
1¼	9	27073	Extra strong
General average		29053	
Iron, Lap-Welded			
2	9	24581	Standard pipe
2	2	34340	Extra strong
General average		29461	

[1] These only are included in averages.

wall, perfection of weld, etc., give rise to variations in bursting strength of sufficient magnitude to render unnecessary any consideration of Birnie's or Clavarino's

condition of head support as shown in Figs. 1 and 2 respectively.

b The relative strengths of steel pipes and tubes, when using Barlow's formula and basing the calculations on average diameter, thickness of wall and ultimate tensile strength of material, are as follows: For butt-welded steel pipe, 73 per cent; for lap-welded steel pipe, 92 per cent; and for seamless steel tubes, approximately 100 per cent. In steel pipe, then, the strength of the butt-weld is about 80 per cent of that of the lap-weld.

c The relative strengths of wrought-iron and steel pipe, from Table 3, are as follows: Butt-welded wrought-iron pipe is 70 per cent as strong as similar butt-welded steel pipe; and lap-welded wrought-iron pipe is 57 per cent as strong as similar lap-welded steel pipe.

41 *Applicability of Barlow's Formula.* Of the five formulae considered in Part I, that by Barlow is the best suited for all ordinary calculations pertaining to the bursting strength of commercial tubes, pipes and cylinders. The theoretical error on the side of safety resulting from its use will generally not exceed the actual combined error on the side of danger when using either Birnie's or Clavarino's formula, due to the ordinary range of variation in the thickness of wall, strength of the material, etc., when applied to the ordinary commercial product. This is true, at least up to the yield point of the material, for any ratio of thickness of wall to outside diameter less than three-tenths. In this respect Barlow's formula is very superior to the common approximate formula which gives errors that are absurdly large on the side of danger for very thick walls (Fig. 4).

42 For all ordinary calculations, then, pertaining to the bursting strength of commercial tubes, pipes and cylinders use Barlow's formula [1] (equation [5])

$$\frac{p}{f}=2\frac{t}{D_1};\quad p=2f\frac{t}{D_1};\quad t=\tfrac{1}{2}D_1\frac{p}{f};\quad f=\tfrac{1}{2}D_1\frac{p}{t}$$

where

D_1 = outside diameter, in.

[1] For certain classes of seamless tubes and cylinders and for critical examinations of welded pipe, where the least thickness of wall, yield point of material, etc., are known with accuracy, and close results are desired, see Clavarino's formulae and Birnie's equations [7] and [10].

t = average thickness of wall, in.
p = internal fluid pressure, lb. per sq. in.
n = safety factor as based on ultimate strength
f = working or safe fiber stress, lb. per sq. in.

$= \frac{40{,}000}{n}$ for butt-welded steel pipe

$= \frac{50{,}000}{n}$ for lap-welded steel pipe

$= \frac{60{,}000}{n}$ for seamless steel tubes

$= \frac{28{,}000}{n}$ for wrought-iron pipe

43 These average values of f are based on the accompanying tables of bursting tests of commercial tubes and pipes. They are intended for substitution in Barlow's formula in case more exact data for the working fiber stress are not at hand.

DISCUSSION

C. M. Sames.[1] About four years ago the writer had occasion to make a considerable number of calculations on the strength of cylinders, and employed C. Bach's formula given in Des Ingenieurs Taschenbuch "Huette." This formula, adopting Professor Stewart's notation, is identical with Clavarino's, excepting that both numerator and denominator of the fraction under the radical sign are divided by ten. Inspection of a numerical table accompanying Bach's formula revealed the fact that it was possible to write an extremely simple expression that would give practically the same values. This expression is

$$t = \frac{0.42\ pD_2}{f - p}$$

and it yields results in no case deviating more than 1.4 per cent from those obtained by using Bach's (Clavarino's) formula for a range of $f = 3p$ to $f = 100p$. This formula was first published in the 1909 edition of an engineering handbook compiled by the writer. Inasmuch as the value (0.3) assumed for Poisson's ratio is at best an approximation, the simplified expression may be used with confidence where the Clavarino formula is called for.

The same expression may be used in place of the Birnie formula by

[1] Editor, Industrial Engineering, New York

substituting the numerical coefficient 0.495 for 0.42 when $f=10p$ to $100p$, and 0.483 when $f=3p$ to $10p$. In the first case the maximum deviation from the values obtained when using Birnie's formula is less than 1 per cent; in the second, less than 1.6 per cent.

The practical accuracy of this simplified formula may be shown by using it to solve Problem 2 in Professor Stewart's paper. Here it is required to determine the fiber stress f in the wall of a cylinder stressed both longitudinally and circumferentially, the outside diameter being 5.5 in., thickness $t=0.25$ in. (making $D_2=5$ in.), and $p=1500$ lb. per sq. in. Substituting these values in the simplified formula

$$0.25=(0.42\times1500\times5)\div(f-1500)$$

from which $0.25f-375=3150$, and $f=14{,}200$ lb. per sq. in. Professor Stewart's method, using his table, gives $f=14{,}200$ lb.; using the Clavarino formula, $f=14{,}093$ lb. The discrepancy here is due to the fact that Professor Stewart used the nearest tabular value instead of interpolating.

T. A. Marsh. Lamé's formula may perhaps be arranged in a more convenient form than that shown in Par. 17, as follows: Using the notations

$S=$ Maximum allowable fiber stress per square inch
$R=$ Outer radius of cylinder, in inches
$r=$ Inner radius of cylinder, in inches
$P=$ Working pressure within the cylinder
$T=R-r$ thickness of the cylinder wall, in inches

$$S=P\frac{R^2+r^2}{R^2-r^2} \quad [11]$$

$$R=r\sqrt{\frac{S+P}{S-P}} \quad [12]$$

$$r=R\sqrt{\frac{S-P}{S+P}} \quad [13]$$

$$P=S\frac{R^2-r^2}{R^2+r^2} \quad [14]$$

$$T=r\left(\sqrt{\frac{S+P}{S-P}}-1\right) \quad [15]$$

$$\frac{R}{r}=\sqrt{\frac{S+P}{S-P}} \quad [16]$$

Using equation [16] the writer at one time prepared a table of ratios of outer to inner radii over a range of allowable stresses and working pressures such as are usually encountered in the design of hydraulic press cylinders. (Table 5.)

In high-pressure work of the nature above referred to, it is evident that metals of low tensile strength have early limitations. Considering equation [16], it is evident that the ratio $\frac{R}{r}$ increases very rapidly as the working pressure approaches the allowable stress S. The ratio becomes infinite when $P=S$ and becomes an imaginary quantity when $P>S$. This means commercially that there is a limiting pressure beyond which it is impossible to design a safe cylinder, and a metal of higher tensile strength must be employed. The cost of materials will define the point beyond which it is economy to resort to better grades of metal.

Sanford A. Moss. For the two tests which Professor Stewart gives in Par. 31, and, in fact, for most cases of commercial tubes and pipes, the ratio of wall thickness to outer diameter is so small that there is no appreciable difference between the various formulae mentioned. For such cases, say for $\frac{t}{D_1}$ up to about 0.03, Professor Stewart's use of Barlow's formula as being best for engineering purposes is undoubtedly justifiable for the reasons which he gives.

For the case of thick cylinders where the ratio $\frac{t}{D_1}$ has a large value there is a great deal to be said, however, in addition to the data which Professor Stewart gives. For such cases he seems to imply that Clavarino's or Birnie's formulae are proper ones to use for the respective cases with and without end heads. According to Guest's maximum shear law, which is now usually accepted, these formulae are considerably in error.

In any case in which there are stresses in two or three directions at once, some criterion must be known for computation, from the various stresses and strains, of an effective stress, which corresponds, so far as yielding is concerned, to the stress in a bar under simple tension in a testing machine.

The theory of elasticity as it has existed for years gives complete information regarding the computation of the stresses and strains, and the only matter in question is the criterion for the effective stress.

TABLE 5 RATIO OUTSIDE RADII TO INSIDE RADII, THICK CYLINDERS

Allowable Stress of Metal per Sq. In. Section, Lb.	Working Pressure in Cylinder, Lb. per Sq. In.												
	1000	1500	2000	2500	3000	3500	4000	4500	5000	5500	6000	6500	7000
2,000	1.732	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
2,500	1.527	2.000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
3,000	1.414	1.732	2.236	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
3,500	1.341	1.581	1.915	2.449	.....	.....	.....	.....	.....	.....	.....	.....	.....
4,000	1.291	1.483	1.732	2.081	2.645	.....	.....	.....	.....	.....	.....	.....	.....
4,500	1.253	1.414	1.612	1.871	2.236	2.828	.....	.....	.....	.....	.....	.....	.....
5,000	1.224	1.362	1.527	1.732	2.000	2.380	3.000	.....	.....	.....	.....	.....	.....
5,500	1.201	1.322	1.464	1.633	1.844	2.121	2.516	3.162	.....	.....	.....	.....	.....
6,000	1.183	1.291	1.414	1.558	1.732	1.949	2.236	2.645	3.316	.....	.....	.....	.....
6,500	.....	1.264	1.374	1.500	1.647	1.825	2.049	2.345	2.768	3.464	.....	.....	.....
7,000	.....	1.243	1.341	1.453	1.581	1.732	1.914	2.144	2.449	2.886	3.605	.....	.....
7,500	.....	1.224	1.314	1.414	1.527	1.658	1.813	2.000	2.236	2.549	3.000	3.741	.....
8,000	.....	1.209	1.291	1.381	1.483	1.599	1.732	1.889	2.081	2.323	2.645	3.109	3.872
8,500	.....	1.194	1.271	1.354	1.446	1.548	1.666	1.802	1.963	2.160	2.408	2.738	3.214
9,000	.....	1.183	1.253	1.330	1.414	1.507	1.612	1.732	1.871	2.035	2.236	2.440	2.828
9,500	.....	.....	1.235	1.306	1.386	1.472	1.566	1.673	1.795	1.936	2.104	2.309	2.569
10,000	.....	.....	1.224	1.291	1.362	1.441	1.527	1.623	1.732	1.856	2.000	2.171	2.380
10,500	.....	.....	1.212	1.274	1.341	1.414	1.493	1.581	1.678	1.789	1.915	2.061	2.236
11,000	.....	.....	1.201	1.260	1.322	1.390	1.464	1.544	1.633	1.732	1.844	1.972	2.121
11,500	.....	.....	1.193	1.247	1.306	1.369	1.437	1.511	1.593	1.683	1.784	1.897	2.027
12,000	.....	.....	1.183	1.235	1.291	1.359	1.414	1.483	1.558	1.640	1.732	1.834	1.949
12,500	.....	.....	.....	1.224	1.277	1.333	1.393	1.457	1.527	1.603	1.687	1.779	1.878
13,000	.....	.....	.....	1.215	1.264	1.318	1.374	1.434	1.500	1.570	1.647	1.732	1.825
13,500	.....	.....	.....	1.206	1.253	1.303	1.357	1.414	1.475	1.541	1.612	1.690	1.775
14,000	.....	.....	.....	1.197	1.243	1.291	1.341	1.395	1.453	1.514	1.581	1.653	1.732
14,500	.....	.....	.....	1.189	1.233	1.279	1.327	1.378	1.432	1.490	1.553	1.620	1.693
15,000	.....	.....	.....	1.183	1.224	1.268	1.314	1.362	1.414	1.469	1.527	1.590	1.658
15,500	.....	.....	.....	1.177	1.216	1.258	1.304	1.348	1.397	1.449	1.504	1.563	1.627
16,000	.....	.....	.....	1.170	1.209	1.249	1.291	1.335	1.381	1.431	1.483	1.538	1.599

In a thick cylinder subject to internal fluid pressure there is a tangential stress which is a maximum at the inner radius, and a radial stress. At the inner radius, the radial stress is equal to the pressure. If the cylinder has heads there is in addition an axial stress, due to the load of the pressure on the inner circle, uniformly distributed over the area given by the difference between the circles.

An old theory of the matter was the "maximum stress" theory. This stated that yielding would begin when the maximum stress in any direction reached the yield point for simple tension, regardless of the existence of the other stresses.

The Lamé formula, cited by Professor Stewart, correctly gives the maximum stress, which is the tangential stress at the inner diameter. Therefore, according to the maximum stress theory, it is the effective stress which must be considered in design work. This theory, however, seems to be wholly abandoned.

Another old theory is the "maximum strain" theory, and it is on this that Clavarino's and Birnie's formulae are based. This states that yielding begins when the strain or deformation in any direction, due to the stress in that direction as well as to the lateral contraction due to stress or stresses at right angles, reaches the value corresponding to yielding due to simple tension.

The simple stress which would produce such a strain is called the true stress and is found by adding to the actual stress, called the apparent stress in this theory, three-tenths of any stress at right angles with proper signs. The factor 0.3 is the value taken for Poisson's ratio.

In a thick cylinder the maximum strain is the tangential strain at the inner diameter. The Birnie formula, which is the maximum strain formula for a thick cylinder without heads is, therefore, obtained by adding to the tangential stress given by the Lamé formula 0.3 of the internal pressure, which is the radial compressive stress.

The Clavarino formula is the maximum strain formula for the case of a cylinder with heads. The axial tension decreases the tangential strain so that the effective stress for a cylinder with heads is less than when there are no heads, according to the maximum strain theory. This is shown by the difference between the Birnie and Clavarino curves in Figs. 3 and 4.

The "maximum shear" theory originated by Guest in England and Mohr in Germany is now usually accepted by engineers who have investigated the matter. This states that yielding begins when the shearing stress reaches a certain value. Hence, when there are

stresses in different directions it is necessary to compute that simple stress which produces an equal maximum shear. It can be shown that when there are two stresses with opposite signs the effective simple stress is the numerical sum of the two values. In the present case the maximum effective stress is at the inner diameter and is found by adding to the tangential tensile stress given by Lamé's formula, the radial compressive stress, given by the internal pressure.

The situation is the same whether there are heads or not, since the axial tension added by the heads is less than the tangential tension and makes no difference in the shear due to the tangential tension and radial compression. The equivalent stress thus computed is greater than that computed by Birnie's formula, since the whole radial stress is added instead of 0.3 of it. It is, of course, greater than Clavarino's formula, since nothing is subtracted on account of the axial tension caused by the heads.

Using Professor Stewart's notation, the effective stress as thus computed on the basis of the maximum shear theory is as follows:

$$\frac{P}{f}=2\frac{t}{D_1}\left(1-\frac{t}{D_1}\right)$$

This is the formula which the writer believes should be used for thick tubes and cylinders instead of the formulae cited by Professor Stewart.

It will be noted that this is the same as Barlow's formula, with the factor $\left(1-\frac{t}{D_1}\right)$, giving a decrease of safe pressure. This formula always gives values lower than the old formulae given by Professor Stewart. As stated, the result is the same with and without the heads, whereas the Clavarino formula gives an increase of safe pressure when there are heads.

THE AUTHOR. There appears to be considerable merit in the approximate form of Clavarino's formula as given by Mr. Sames because of its simplicity when used for making calculations.

I infer from Mr. Marsh's discussion that he prefers to express Lamé's formula in terms of the outside and inside radii of the cylinder. The formula as expressed in the paper is in terms of the outside and inside diameters because for commercial steel pipe and tubes it is customary to state the diameters or one diameter and wall thickness.

I have noted with considerable interest what Mr. Moss states regarding the application of Guest's maximum shear theory for

thick-walled tubes and pipe. I cannot accept his conclusions for the following reasons:

a A perfect formula for the strength of thick-walled tubes and pipe would apply equally well to those having thin walls. In the absence of specific reasons to the contrary it is reasonable to suppose that the more perfect of two theoretical formulae for the strength of thick-walled tubes would apply equally as well to thin-walled tubes as would the less perfect formula.

b Applying Clavarino's formula, based on maximum strain theory, and the formula given by Mr. Moss, based on maximum shear theory, to the conditions of the summary of results of the 474 tests given in Par. 31 of this paper, we get calculated values of internal fluid pressure at the instant of reaching yield point of innermost fibers of pipe as follows:

FOR 10-IN. PIPE

Average of 195 tests, actual fluid pressure in lb. per sq. in	1435
By Clavarino's formula, fluid pressure in lb. per sq. in	1490
By Mr. Moss' formula, fluid pressure in lb. per sq. in	1251
Error by Clavarino's formula, per cent	3.8
Error by Mr. Moss' formula, per cent	12.8

FOR 12-IN. PIPE

Average of 279 tests, actual fluid pressure in lb. per sq. in	1195
By Clavarino's formula, fluid pressure in lb. per sq. in	1191
By Mr. Moss' formula, fluid pressure in lb. per sq. in	1000
Error by Clavarino's formula, per cent	0.3
Error by Mr. Moss' formula, per cent	16.3

No. 1353

ON THE CONTROL OF SURGES IN WATER CONDUITS

By W. F. Durand, Stanford University P. O., Cal.

Member of the Society

The literature of the problem of the surge chamber as a means of control of surges in water conduits has recently been enriched by two important papers presented before the Society, one by Raymond D. Johnson[1] and another by William F. Uhl,[2] in the latter case the treatment of the surge chamber being incidental to its relation to the general problem of speed regulation.

2 The attention of the writer has been directed recently to the examination of certain cases of surge-chamber design in which funnel or tapering forms rather than cylindrical forms are indicated by the environment and conditions of the problem. This has led to a re-examination of the various methods of treatment available, and to the development of a method by which, without prohibitive labor, the approximations of the various analytical and empirical methods may be avoided and a chamber determined of any desired degree of taper and suited to the development of any desired program as to rate of acceleration and amplitude of surge.

3 In connection with Fig. 1, let us assume the following notations:

v = velocity in general in conduit AB, assumed here of uniform cross-section area
v_1 = initial velocity for period under consideration
v_2 = velocity when movement of water level reaches first maximum
v_3 = velocity for ultimate steady conditions
v' = velocity such that $v'A$ = volume of water demanded by wheels under governor control for constant power during period under consideration

[1] Trans. Am. Soc. M. E., vol. 30, p. 443.
[2] The Journal Am. Soc. M. E., February 1911, p. 169.

Presented at the San Francisco Meeting, June 1911 and the Spring Meeting, Cleveland, 1912, of The American Society of Mechanical Engineers.

A = cross-section area of conduit
F = cross-section area of surge chamber at water level
$n = \frac{F}{A}$
L = length of conduit AB
y = distance of water level in surge chamber from initial position for steady motion with velocity v_1
Y = distance of water level in surge chamber from static level
$a = \frac{L}{g}$
b = factor such that bv^2 = sum of friction and velocity heads at B; then $Y = y + bv_1^2$ for demanded load
and
$Y = -y + bv_1^2$ for rejected load
H = gross head at wheels
h = total head involved in friction and acceleration
C = coefficient in Chezy formula[1]
r = hydraulic mean radius of conduit

4 In practice h will be of the form

$$h = fv^2 + h_0 + a\frac{dv}{dt}$$

where f = the value of $\frac{L}{C^2r}$ in the Chezy formula, v = the velocity in AB, h_0 = the sensibly constant friction loss for the part of the conduit or penstock lying between the surge chamber and the wheels, computed by the same formula and with a velocity corresponding sensibly to v_3 in AB and $a\frac{dv}{dt}$ = acceleration head. For rejected load $a\frac{dv}{dt}$ becomes negative and is therefore subtractive from the sum of $fv^2 + h_0$.

5 It should be noted that we shall here use the term "surge" not in the sense of the total or maximum movement of water during a period of changing load, but rather as the maximum distance by which the water level passes beyond its final location for steady conditions with the corresponding ultimate velocity.

6 In Appendix No. 1 it is shown that the differential equations for the movement of the water in the conduit and surge chamber are as follows:

[1]Friction head = $\frac{Lv^2}{C^2r}$

Demanded load.....

$$a\frac{dv}{dt} = y - b\,(v^2 - v_1^2) \quad \text{.....[1]}$$

$$n\frac{dy}{dt} = (v' - v) \quad \text{.....[2]}$$

Rejected load.....

$$-a\frac{dv}{dt} = y - b\,(v_1^2 - v^2) \quad \text{.....[3]}$$

$$n\frac{dy}{dt} = (v - v') \quad \text{.....[4]}$$

7 For constant power during the period of change, equal to that under steady conditions with the velocity v_3, we shall also have:

$$v'\,(H - h) = v_3\,(H - h_3) \quad \text{.....[5]}$$

or

$$v' = v_3\,\frac{H - h_3}{H - h} \quad \text{.....[6]}$$

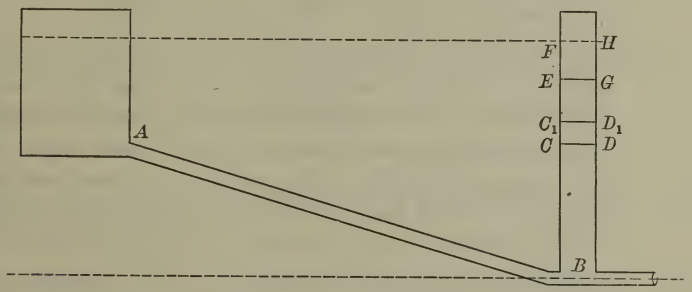


FIG. 1 DIAGRAM SHOWING ARRANGEMENT OF CONDUIT AND SURGE CHAMBER

EG denotes level for steady motion with velocity v_1
C_1D_1 denotes level for steady motion with velocity v
CD denotes actual level with velocity v
$EC = y$
$FC = Y$

8 These equations in effect represent a damped oscillation in which the damping force varies with the square of the velocity. In this form they have not been integrated, nor do they seem susceptible of integration, at least in terms of present known functions.

9 Under these circumstances, several courses are open as follows:

a The omission of the frictional or damping term and of the influence due to governor control. This is equivalent to taking $b = o$ and $v' = v_3$. This simplifies the problem to such a degree that it is readily treated by

exact mathematical methods. Such results have been developed many times and discussed by various authors. The assumed conditions are, however, so far removed from the actual that the results are of small value except as a first step in the approach to the real problem.

b The substitution of a damping force varying as the first power of the velocity instead of as the square. This with suitable assumptions regarding the influence due to governor control makes possible the integration of the equations and renders the treatment similar to that for certain types of electrical oscillations. This substitution lies at the foundation of the masterly analytical treatment of the problem by certain eminent European engineers.[1] At the best, however, this method is approximate and must needs be under control by reference to some more exact method in order to know the order of error involved or to determine values of arbitrary constants.

c The development of approximate and empirical equations upon the various assumptions involving more or less simplification of the original equations, and involving the use of empirical coefficients derived from actual cases by means of method *d*. The paper by Johnson previously referred to, with the accompanying discussion, furnishes a large number of such formulae developed either by the original author or by the authors of the discussion, and by means of which most problems within the field of control may be very satisfactorily treated. By the field of control is here meant the field over which are scattered the actual cases worked out by more exact methods and the results of which have been used to check the coefficients and results in general for the various approximate equations.

d The equations may be treated by approximate numerical integration in a step-by-step trial and error process. It will be obvious that by an appropriate procedure the equations may be treated by this general method whether n is constant or variable so long as the proposed form of chamber is specified.

[1] Prasil, Wasserschlossprobleme—Schweizer Bauzeitung—Band 52 Nr. 21, 23, 24, 25.

10 In the methods of treatment indicated under *a*, *b* and *c*, it is of course, assumed that n is a constant. The more general assumption that n is a variable, that is, that the chamber may be of conical or vase-like form, would require the expression of n as a function of y. This would result in the general case in a further complication which would render still more difficult or impossible the treatment of the problem on the assumptions made in *a*, *b* and *c*.

11 In the literature of the present subject many illustrations[1] of method *d* for constant n will be found in the annals of the Society as well as in other engineering and scientific works. The writer has done, perhaps, a fair share of the tedious work involved in the application of this method, and it was in a measure as a revolt against this

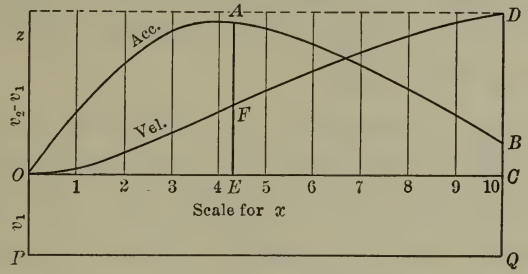


Fig. 2 Diagram showing Relation of Acceleration and Velocity Curves on Time Axis

serious expense of time, that the method which forms the subject of the present paper was developed.

12 In all of the preceding methods whether by analytical approximation or by numerical integration, the method of approach to the problem is as follows: Given a specified or proposed surge chamber, what will happen? Let us reverse the order and state the question thus: Given a program for the water, required a surge chamber to produce it.

13 To develop the possibilities of this viewpoint, let us fix in advance the character of the acceleration of the water in the conduit as indicated by the curve *OAB* in Fig. 2, in which diagram time is laid off on the x-axis, but without as yet any determination of scale; and acceleration is laid off on the z-axis without as yet any determination of scale.

[1] The Surge Tank in Water-Power Plants, R. D. Johnson, Trans. Am. Soc. M. E., vol. 30, p. 443.

14 The curve OAB is then taken as the proposed history of the acceleration up to the time when the water reaches its highest or lowest level, according as the case is for rejected or demanded load. In order to fix the ideas, we shall for the present assume the case of demanded load and a falling surface of water. Then the acceleration curve OAB and the time OC both relate to the period from the start to the attainment of a maximum drop of water surface or to a maximum value of y. The plotting of this curve does, therefore, nothing more than to fix in advance the general type or form of the acceleration curve, leaving all questions of actual numerical values for later determination. Methods for the determination of the scale factors and the numerical values will be found in Appendix No. 2. In order not to interrupt the continuity of development of the general method, the question of scales may be waived for the moment and we may assume OAB to represent definitely the time history of the acceleration $\frac{dv}{dt}$.

15 By means of a planimeter, or by arithmetical means, or by an integraph, if such an instrument should be available, the first integral curve of OAB is readily found as shown by OD. This will give the time history of the change in velocity from v_1 to v_2. Lay off v_1 as OP and draw PQ. Then from PQ as an axis the curve OD will give the complete time history of the velocity in its growth from $PO = v_1$ to $QD = v_2$.

16 Now, writing equations [1], [2] and [5], we have

$$a\frac{dv}{dt} = y - b(v^2 - v_1^2) \quad \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

$$n\frac{dy}{dt} = (v' - v) \quad \ldots\ldots\ldots\ldots\ldots\ldots [8]$$

$$v' = v_2\frac{H - h_2}{H - h} \quad \ldots\ldots\ldots\ldots\ldots\ldots [9]$$

In equation [7], it is readily seen that we now know all quantities except y. Hence, by direct substitution, we find y for a series of values of the time, and if desired we may plot the result as a curve of y on time.

17 This curve is indicated in Fig. 3 by OAB. Let CA denote the level for steady conditions with the final velocity v_2, then $OC = b(v_2^2 - v_1^2)$, while CD denotes the extent of the surge or drop below CA, the final level for steady conditions. We next proceed to differentiate

the curve OAB. This may most easily be done by numerical methods as given in Appendix No. 2. At any given point, or value of the time, this will give the value of $\frac{dy}{dt}$. We may also find v' from [9], and thus in [8] we have all quantities known except n, which may be found by the solution of this equation.

18 The time histories of all quantities, $\frac{dv}{dt}$, v, y and n, are now known. We may, as desired, plot $a\frac{dv}{dt}$, v and n on y and in particular may reduce n to the diameter of the chamber and plot on y, thus showing for the chamber the form and dimensions required. In case the influence of governor control is omitted, we may put $v' = v_3$ in [8] and discard [9]. This will simplify the operations to a considerable degree.

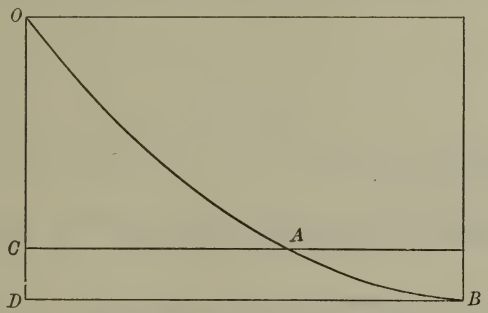


FIG. 3 DIAGRAM SHOWING MOVEMENT OF WATER LEVEL ON TIME AXIS

19 The time required for the complete treatment of a given case by this method might be from two to four hours, depending on the number of points found, the degree of accuracy required, etc. The aid of a slide rule of the Fuller or Thatcher form and giving a fifth place by estimation is assumed.

20 After a given case has been worked through for a single value of the time, a second case with a different value of the time and the same acceleration curve, except for slight modifications near the end, may be worked through with considerable saving of time. In this manner, a series of forms of the chamber may readily be derived from a single base or type curve of acceleration.

21 After the area of the acceleration curve has been found, the areas of the chamber at the top and near the bottom, or at the top near the bottom and at one or two intermediate points may quickly be found, and independently of the detailed determination of the

curve of y as described in Par. 16. In this manner the general character of the chamber may be readily foreseen in advance of the complete and detailed determination of its form. The resulting value of y_{max} and hence the amount of surge may also be found immediately and in advance of the detailed determination of the y curve or of the form of the chamber. The general characteristics of the acceleration curve may also be readily adjusted to produce a chamber of any desired general character of form and size.

22 In Appendix No. 3 will be found further discussion of these details of the operative program, together with the numerical work for a typical case, the latter indicating in detail the various steps. A study of this form of computation together with the fundamental equations, will better serve to acquaint the reader with the operative details of the method than further descriptive treatment.

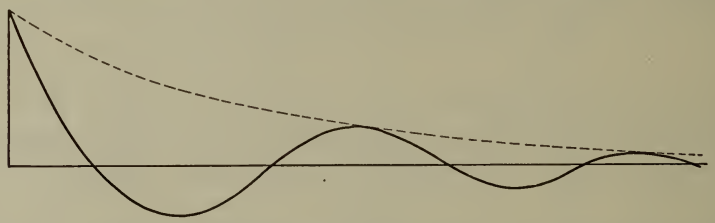


FIG. 4 DIAGRAM ILLUSTRATING OSCILLATORY AND NON-OSCILLATORY MOVEMENTS

NON-OSCILLATORY MOVEMENT

23 One special case merits particular attention. The movement of the water surface has been referred to as that of a damped oscillation. The peculiar characteristic of such a movement lies in the fact that it involves a continuous and indefinite approach to a final level or condition or axis by means of an indefinite series of swings or oscillations of decreasing amplitude and lying on either side of the final level, or axis, and as represented by the full line in Fig. 4.

24 Now the limiting case of such a movement is that of a single swing or indefinite approach to the final level or condition or axis, but without passing beyond, and lying wholly on one side of such a final goal. Such a movement is indicated by the dotted line in Fig. 4. This condition has its direct analogy in certain types of electrical non-oscillatory discharge.

25 Now in the case of a hydraulic conduit with surge chamber, the realization of a non-oscillatory movement of the water in the chamber may well be considered as a most desirable ideal in case the

needed dimensions are practicable. The dimensions and form of such a chamber are readily determined by the choice of an appropriate form of acceleration curve combined with a suitable time allowance.

26 It is clear when the water reaches its final level that in all cases we must have simultaneously the velocity $\frac{dy}{dt}$ and the acceleration $\frac{dv}{dt}$ equal to zero. This results from the form of the y curve which is here coincident with the time axis. If this final condition is to be realized by a non-oscillatory movement, then the one swing of the water surface which represents the passage from the initial level to its maximum drop and the simultaneous change of velocity from v_1 to v_3 must be conditioned by a suitable form of acceleration curve. Obviously, this will be a curve of the form indicated in Fig. 5, where

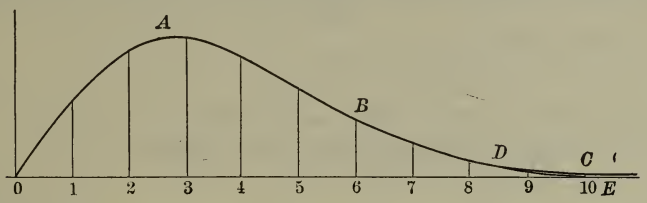


FIG. 5 DIAGRAM SHOWING TYPE OF ACCELERATION CURVE FOR NON-OSCILLATORY MOVEMENT

$OABC$ indefinitely approaches x and the value of the acceleration indefinitely decreases as time increases. Theoretically, we should realize the final conditions only after an infinite time.

27 Practically two courses are open:

a A form of curve $OABC$ where the terminal point C lies near but not quite on the x-axis is assumed. In other words, a finite part of the indefinitely extended curve is used implying for the conditions when y reaches its first maximum value, a slight residual acceleration, and hence a slight surge or movement of the water surface beyond its level for steady conditions. By taking C near the axis this surge may be reduced to an entirely insignificant value, and the resulting movement thus made sensibly non-oscillatory. In such a case the mode of procedure is the same as for the general case.

b A form of curve $OABDE$ in which the curve is brought down tangent to x within a finite distance is assumed.

28 In the general case with an acceleration curve of the form shown in Fig. 2, or *OABC*, Fig. 5, it is shown in Appendix No. 2 that there is but one particular value of the time which will fit the various conditions imposed, and the time thus determined is therefore that required for the first swing to a maximum value of y. In case, however, the curve is assumed of the form *OABDE*, it is shown that the time is not determined by the conditions thus imposed, and in consequence we are free to choose any value of the time whatever, as long as it does not fall below a certain limiting minimum. For each value of the time which may thus be taken, a corresponding chamber will result suitable for the realization of the proposed program of acceleration, non-oscillatory movement of the water, etc., and all within the assumed time. In this manner a series of chambers may be developed from one assumed base curve of acceleration, the required work for all cases after the first being comparatively small. An illustrative case with further discussion of operative details will be found in Appendix No. 3.

RELATIONS BETWEEN FORM OF ACCELERATION CURVE AND CHARACTERISTICS OF CHAMBER

29 Regarding the selection of a form of acceleration curve which will correspond to any proposed general form of chamber a very little experience with the treatment of such problems serves to show the general type of acceleration curve produced by chambers of cylindrical or tapering form. The value of the acceleration starting at zero mounts rapidly to a maximum during the early part of the total period, and then falls off more gradually to its value for $v = v_2$, either zero or some relatively small value, as shown in Figs. 2 and 5, according as it is intended to have the movement non-oscillatory or to permit a small surge movement. It may also be readily seen in what manner the form of the curve should be modified in order to produce any desired change in the amount of surge, the time required or the general form of the resulting chamber.

30 A further important question in the case of non-oscillatory movement refers to the relation between the change in the time allowed and the resulting change in the form of the chamber, the form of the acceleration curve remaining the same.

31 In Appendix No. 2 will be found a detailed discussion of these points including the development of such relations as will

provide for the ready adjustment of a curve of acceleration suited to develop any general form of chamber as desired, and to secure any specified program as regards the movement of the water, permissible extent of surge, time required, etc.

THE RELATION BETWEEN DEMANDED AND REJECTED LOAD

32 The general discussion of this method of treatment has assumed the condition of demanded load where any differences were involved. It is evident that precisely the same methods may be applied to the case of rejected load. The latter investigation may be of special importance in connection with the permissible height of surge in case a moderate surge may be permitted. It will scarcely be necessary to discuss this case in detail, or to develop the special forms which the various equations will take.

33 An examination of the various equations for this case will indicate that for a given program of acceleration, etc., a chamber of different form will be required than for the case of demanded load. Or otherwise with a given chamber, the program of water movement, extent of surge, etc., will be somewhat different from the case with demanded load.

34 With a cylindrical form of chamber the surge for rejected load between the same limits of velocity will be more severe than for demanded load. With tapering forms where for demanded load the initial drop in water level and rate of development of acceleration head are slow at first, while for rejected load they are rapid at first, the two surges may become substantially equal or may differ in such degree as can be determined only by detailed examination.

35 If for a chamber developed for the case of demanded load it is desired to determine the detailed results in the case of rejected load, there seems to be no more direct method than the regular solution of the equations for this case by means of approximate integration and trial and error adjustment.

SOURCE AND CHARACTER OF ERRORS

36 The degree of accuracy which may be expected in the solution of problems of this character, together with the sources of uncertainty or error merit brief notice. Errors in the final results may arise from four chief causes:

a The failure of the general equations properly to include all the factors or conditions of the problem.

b The adoption of approximate methods of treatment involving the omission of various factors or the simplification of their relations to the problems, all for the purpose of bringing the equations or the resulting conditions within the range of ready mathematical treatment.

c Uncertainty regarding the values to be ascribed to various empirical constants.

d The inaccuracies of numerical computation.

37 In the case of the present problem, errors of class *a* may arise from the failure of the governor to work perfectly, and from a slight change in the efficiency of the wheels with change in effective head and hence in spouting velocity. These conditions will affect the virtual velocity v' and hence in any given case the actual values of v' may not be equal exactly to those given by equation [6].

38 In no actual case, furthermore, does the load remain perfectly constant. In many cases it may remain sensibly constant, while in some the governor will be continually chasing the conditions developed by the load, but never quite overtaking or controlling them. Assuming, however, for the sake of definiteness of condition that the load is uniform and that the velocity v' is properly given by [6], it appears that all remaining conditions with the exceptions noted in Par. 39 are properly represented in the general equations.

39 The influence of acoustic waves and their time element are not included in the fundamental equations as developed. There is good reason for such omission in the fact that the time element introduced by the acoustic wave is small compared with the time involved in the rise and fall of water in a surge chamber, and no sensible error will result from such omission in dealing with surge chamber problems. On the other hand, as is well known, the time element introduced by the acoustic wave must be considered in the general problem of governor control.

40 Errors of class *b* are necessarily present in all analytical treatments of the present problem. The present method eliminates errors of this class.

41 Errors of class *c* in the proposed method of treatment depend solely on the value of the coefficient for loss of head due to friction. It will be noted that the coefficient *b* in the general equations is the only one which involves empirical constants of this character.

42 Errors of class *d* are wholly under control. They may be reduced to any desired limit by a suitable adjustment of the program of numerical computation.

43 It follows in the proposed method of treatment, assuming the conditions of the problem to be properly represented by the general equations, that the only source of unavoidable error lies in the coefficient for friction loss. The results therefore, when derived by the aid of adequate methods of numerical computation, must represent the consequences of the assumed conditions with the same degree of accuracy as that with which we may compute the final effective head, the horsepower with an assumed wheel efficiency, or any other simple result for which the friction head is a determining factor.

APPENDIX No. 1

GENERAL EQUATIONS OF MOTION FOR THE CASE OF A CONDUIT MADE UP OF SECTIONS OF DIFFERENT SIZES AND HENCE CONTAINING COLUMNS OF WATER MOVING AT DIFFERENT VELOCITIES

44 The general equations for the motion of the water in a pipe line of uniform cross-section under the influence of a surge chamber have been so frequently derived and are so well known in the literature of the subject that their derivation as a special case may be omitted.

45 It not infrequently occurs, however, that the conduit as a whole is made up of sections of different sizes and lengths and hence at any instant contains columns of water of varying weights and moving with different velocities. The usual equations of motion which are based on an assumed uniform section, and hence uniformity of velocity throughout the conduit, cannot be used for this case. It therefore becomes of practical importance to determine the more general equations for this case.

46 Referring to Fig. 6 we shall use the following notation:

A, A_b, A_c to denote areas of sections a, b, c

u, v_b, v_c to denote velocities in sections a, b, c

It will be noted that the conditions in section a are to be taken as standard, and hence the area and velocity for this section are denoted by A and u without a special subscript.

u_1 = initial velocity in section a

u_2 = velocity in section a when movement of water level reaches first maximum

u_3 = velocity in section a for ultimate steady conditions

m = ratio between velocity in any section and that in section a, that is

$$v_b = m_b u$$

$$v_c = m_c u, \text{ etc.}$$

Hence, we shall have

$$A_b = \frac{A}{m_b}$$

$$A_c = \frac{A}{m_c}$$

V = volume of 1 lb. of water

σ = density

C = coefficient in Chezy formula

r = hydraulic mean radius

u' = velocity in section a which would carry just the volume of water demanded by the wheels when under governor control during the period under consideration

h = total lost head between intake of conduit and wheels

Then h = friction head + acceleration head

H = gross head at W

F = area of surge chamber at water level

$$n = \frac{F}{A}$$

y = movement of water level in surge chamber from position for steady motion with velocity u_1

Y = movement of water level in surge chamber from static level

47 To fix the ideas we shall assume the condition of demanded load and hence of increasing velocity, and write the general equation of work-energy for 1 lb. of water entering the conduit at B and passing out at E.[1] We shall take

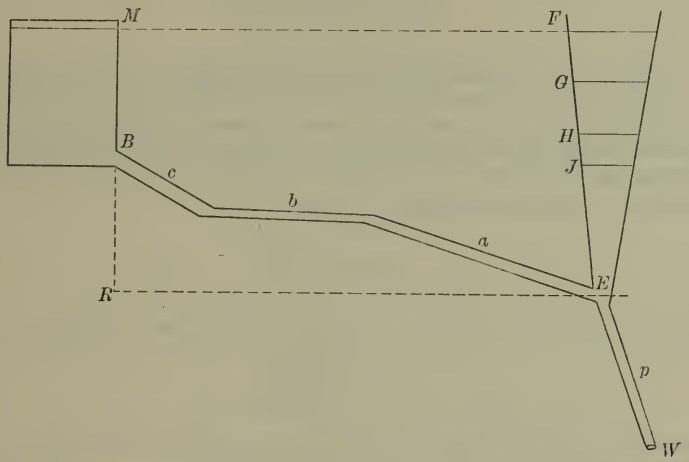


FIG. 6 DIAGRAM SHOWING CONDUIT LINE OF MIXED SIZES WITH SURGE CHAMBER

F denotes level for static conditions
G denotes level for steady conditions with velocity u_1 in section *a*
H denotes level for steady conditions with velocity u in section *a*
J denotes level for actual conditions with velocity u in section *a*

the datum level at E since only the parts of the system lying above this datum are directly involved. At the point B we shall have the following items of work-energy inflow:

$$\frac{v_c^2}{2g} = \text{kinetic energy}$$

$$\text{Head } BR = \text{potential energy}$$

$$P_B V = \frac{P_B}{\sigma} = \text{work done on entering pound of water}$$

These constitute the total work-energy input at B.

[1] The influence due to entrance head at the intake of the conduit is omitted as usually too small to be of significance.

48 Corresponding and equal to this total input, we have a threefold subdivision as follows:

a A part is represented by the work required to overcome frictional resistance in the conduit.

b A part is represented by increase in the kinetic energy of the water in the conduit which is undergoing continuous acceleration and hence is increasing in energy content.

c The remainder is represented by the work-energy output at the point *E*.

Taking first the work required to overcome the skin friction in the conduit, we have for any section of length L, hydraulic radius r, coefficient C and velocity v. Then

$$\text{loss of head}^1 = \frac{v^2 L}{C^2 r}$$

49 This is commonly called loss of head. It is readily seen that in effect it is the measure of the work required to overcome the frictional resistance throughout a conduit of length L per lb. of water entering or leaving, or in general per lb. of water flowing past any given section. In the present case all quantities are taken per lb. of water so that for any section in general, we shall have for the work expended in friction

$$\text{work} = \frac{v^2 L}{C^2 r}$$

For the entire conduit made up of sections of mixed sizes, we shall have

$$\text{work} = \Sigma \left(\frac{v^2 L}{C^2 r} \right)$$

where Σ denotes simply the summation of the series of partial amounts for each individual section of the total conduit. But, in general $v = mu$ and hence

$$\text{work} = \Sigma \left(\frac{v^2 L}{C^2 r} \right) = u^2 \Sigma \left(\frac{m^2 L}{C^2 r} \right)$$

50 We have next for the kinetic energy in any section of the conduit

$$\text{energy} = K = \frac{\sigma A L v^2}{2g}$$

Then in time dt this is subject to change (in the present case increase) by the amount

$$dK = \frac{\sigma A L v}{g} \times \frac{dv}{dt}\, dt$$

Now for 1 lb. to enter at B or to pass E or in general to pass any given point, the time is

$$dt = \frac{1}{\sigma A v}$$

[1] This follows directly from the Chezy formula.

Substituting this value of dt in the above value of dK we have

$$dK = \frac{Ldv}{gdt}$$

Hence the total amount of energy absorbed in the acceleration of the contents of the entire conduit will be

$$\Sigma dK = \Sigma \left(\frac{Ldv}{gdt} \right)$$

But since in general $v = mu$, we have

$$\frac{dv}{dt} = m \frac{du}{dt}$$

and

$$\Sigma dK = \Sigma \left(\frac{Ldv}{gdt} \right) = \frac{du}{dt} \Sigma \left(\frac{mL}{g} \right)$$

51 Finally, at the point of exit E, we have the following items of work-energy output

$$\frac{u^2}{2g} = \text{kinetic energy}$$

$$P_E V = \frac{P_E}{\sigma} = \text{work done on outgoing lb. of water}$$

Hence, collecting and equating we have

$$\frac{v_c^2}{2g} + BR + \frac{P_B}{\sigma} = u^2 \Sigma \left(\frac{m^2 L}{C^2 r} \right) + \Sigma \left(\frac{mL}{g} \right) \frac{du}{dt} + \frac{u^2}{2g} + \frac{P_E}{\sigma} \quad \ldots\ldots [10]$$

But

$$\Sigma \frac{v^2 L}{C^2 r} = u^2 \Sigma \left(\frac{m^2 L}{C^2 r} \right) = \text{total friction head}$$

Hence

$$u^2 \left[\Sigma \left(\frac{m^2 L}{C^2 r} \right) + \frac{1}{2g} \right] = FH$$

Also

$$\frac{P_E}{\sigma} = JE$$

Hence [10] is eqivalent to

$$MR = FH + JE + \Sigma \left(\frac{mL}{g} \right) \frac{du}{dt}$$

or

$$\Sigma \left(\frac{mL}{g} \right) \frac{du}{dt} = MR - (FH + JE) = HJ \quad \ldots\ldots [11]$$

But

$$HJ = y - GH$$

and

$$GH = FH - FG = \begin{cases} \text{difference of total friction and velocity} \\ \text{heads for velocities } u_1 \text{ and } u \end{cases}$$

Hence, we have

$$GH = \left[\Sigma\left(\frac{m^2L}{C^2r}\right) + \frac{1}{2g}\right](u^2 - u_1^2)$$

and [11] becomes

$$\Sigma\left(\frac{mL}{g}\right)\frac{du}{dt} = y - \left[\Sigma\left(\frac{m^2L}{C^2r}\right) + \frac{1}{2g}\right](u^2 - u_1^2) \quad \ldots\ldots [12]$$

It thus appears that in such a conduit with mixed sizes the virtual or equivalent length in the term involving acceleration is ΣmL while in the term involving friction the virtual term $\frac{L}{C^2r}$ is $\Sigma\left(\frac{m^2L}{C_2r}\right)$

52 In accordance with the preceding notation, the amount of water required at any given instant during the period of change is $u'A$. The amount which is coming through section a of the conduit is uA. Hence, assuming $u' > u$ the remainder must come from the surge chamber. In general, we shall have

$$u'A = uA + F\frac{dy}{dt}$$

or

$$\frac{Fdy}{Adt} = n\frac{dy}{dt} = (u' - u) \quad \ldots\ldots [13]$$

Also h = friction head plus acceleration head or

$$h = \Sigma\left(\frac{m^2L}{C^2r}\right)u^2 + \Sigma\left(\frac{mL}{g}\right)\frac{du}{dt} + h_0 \quad \ldots\ldots [13]_1$$

where h_0 represents the sensibly constant friction loss in the section of the conduit between the surge chamber and the wheels, and which may be based on a sensibly uniform velocity corresponding to u_3 in section a. Frequently this term h_0 will vanish. Then for constant power during the period under consideration we shall have

$$u'A\,(H - h) = u_3A\,(H - h_3)$$

Hence

$$u' = u_3\frac{(H - h_3)}{(H - h)} \quad \ldots\ldots [14]$$

53 It is evident that the total friction head as a part of h and h_2 must be found by summing the friction heads for the various parts of the entire line including the penstock p. It should also be noted that the pipe p is carrying the total volume of water while the conduit abc is carrying only a part. The velocities in these various sections must therefore be taken accordingly. No sensible error will be involved if the velocity in p is taken as uniform and of such value as to correspond to u_3 in section a. Rewriting [12], [13] and [14] to bring them together, we have

$$\Sigma\left(\frac{mL}{g}\right)\frac{du}{dt} = y - \left[\Sigma\left(\frac{m^2L}{C^2r}\right) + \frac{1}{2g}\right](u^2 - u_1^2) \quad \ldots\ldots [15]$$

$$n\frac{dy}{dt} = \left(u' - u\right) \quad \ldots\ldots [16]$$

$$u' = u_3\frac{(H - h_3)}{(H - h)} \quad \ldots\ldots [17]$$

54 If the conduit is of uniform section then m becomes 1 in all cases and putting L the length of the conduit for ΣmL and Σm^2L and writing now v instead of u, we have

$$\frac{L}{g}\frac{dv}{dt} = y - \left(\frac{L}{C^2r} + \frac{1}{2g}\right)\left(v^2 - v_1^2\right) \quad \ldots\ldots [18]$$

55 Equations [16] and [17] remain the same in form for this case as for the more general. If we neglect the influence due to governor control and assume that the flow of water is uniform in volume and equal to u_3A then u' becomes u_3 and equation [17] may be discarded.

56 A comparison of equations [15] and [18] shows, as might be expected, that the equation for a conduit with mixed sizes is similar in form to that with uniform size, but with different coefficients for the terms involving acceleration and friction. It also appears that by using the velocity and acceleration in some one section as standard, the case with mixed sizes may be thus reduced to an equivalent or virtual case with uniform size.

57 The preceding various equations in the form given apply to the case of demanded load. For the case of rejected load the modifications are readily made. The resulting equations are

$$-\Sigma\left(\frac{mL}{g}\right)\frac{du}{dt} = y - \left[\Sigma\left(\frac{m^2L}{C^2r}\right) + \frac{1}{2g}\right](u_1^2 - u^2) \quad \ldots\ldots [19]$$

$$n\frac{dy}{dt} = (u - u') \quad \ldots\ldots [20]$$

$$u' = u_3\frac{(H - h_3)}{(H - h)} \quad \ldots\ldots [21]$$

$$h = \Sigma\left(\frac{m^2}{C^2}\frac{L}{r}\right)u^2 + \Sigma\left(\frac{mL}{g}\right)\frac{du}{dt} + h_0 \quad \ldots\ldots [21]_1$$

APPENDIX No. 2

THE DERIVATION OF SCALE RATIOS AND NUMERICAL RELATIONS FOR ANY GIVEN ACCELERATION CURVE

58 Referring to Fig. 2, the following notation will be used.

x to denote the abscissa
z to denote the ordinate
α to denote the scale ratio for z
β to denote the scale ratio for x
R to denote the area of acceleration curve $OABC$

Then for the acceleration corresponding to any point on the curve, we have

$$\frac{dv}{dt} = \alpha z$$

For the time, we have

$$T = \beta x$$

or otherwise β is the value of the time interval or unit adopted for x.

59 From equations [1], [2] and [6] we have

$$y = a\frac{dv}{dt} + b\,(v^2 - v_1^2) \quad \ldots\ldots [22]$$

$$\frac{dy}{dt} = \frac{v' - v}{n} \quad \ldots\ldots [23]$$

$$v' = v_2\frac{H - h_2}{H - h} \quad \ldots\ldots [24]$$

Differentiating [22], we have

$$\frac{dy}{dt} = a\frac{d^2v}{dt^2} + 2bv\frac{dv}{dt} \quad \ldots\ldots [25]$$

Whence from [23] and [25]

$$\frac{v' - v}{n} = a\frac{d^2v}{dt^2} + 2bv\frac{dv}{dt} \quad \ldots\ldots [26]$$

60 At the end of the period under consideration both v' and v become v_2. Hence $v' - v = 0$ and we must have as a necessary condition at the end of such period

$$a\frac{d^2v}{dt^2} = -2bv_2\frac{dv}{dt} \quad \ldots\ldots [27]$$

But referring to the assumed curve of acceleration, we have

$$\left.\begin{aligned} \frac{dv}{dt} &= \alpha z \\ \text{and} \qquad \frac{d^2v}{dt^2} &= \alpha \frac{dz}{dt} = \frac{\alpha}{\beta}\frac{dz}{dx} \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots\ldots\ldots [28]$$

Hence, substituting these values in [27], we have

$$\frac{a\alpha}{\beta}\frac{dz}{dx} = -2bv_2\alpha z$$

Whence

$$\frac{\frac{dz}{dx}}{z} = -\frac{2bv_2\beta}{a}$$

Putting $\frac{dz}{dx} = m$ and solving for β we have

$$\beta = \frac{-am}{2bv_2 z} \quad \ldots\ldots\ldots\ldots\ldots\ldots [29]$$

61 It thus appears that the general equations involve a relation which takes such form where y becomes a maximum at the end of the period (and hence at the terminal end of the acceleration curve), as to provide for the determination of the time scale β and hence of the time T which will be required on the program resulting from the acceleration curve assumed. In other words, the curve as assumed, in conjunction with the constants a, b and the final velocity v_2 will admit of but one value of the time, and hence one value of the time scale β as determined by [29].

62 Again from the relations involved we have

$$\alpha\beta R = \text{total change in velocity} = (v_2 - v_1)$$

whence

$$\alpha\beta = \frac{v_2 - v_1}{R} \quad \ldots\ldots\ldots\ldots\ldots\ldots [30]$$

and

$$\alpha = \frac{v_2 - v_1}{R\beta} \quad \ldots\ldots\ldots\ldots\ldots\ldots [31]$$

It remains to determine v_2. The actual relations of v_2 and h_2 are, however, implicit, and hence a value of v_2 must be found by trial which will result in satisfying equations [29] and [30] and also the following:

$$h_2 = fv_2^2 + h_0 + a\frac{dv}{dt}$$

$$\frac{dv}{dt} = \alpha z$$

$$v_2 = \frac{v_3(H - h_3)}{H - h_3}$$

For demanded load, v_2 will slightly exceed v_3. Often the difference will be too small to be of significance. In any case, the value is readily found by trial substitution in the above equations.

63 Likewise in general, change of velocity for any period of time equals area of corresponding part of acceleration curve multiplied by $\alpha\beta$ or

$$\Delta v = \frac{v_2 - v_1}{R} \int z dx \qquad [32]$$

That is, at any point as E the change of velocity will equal $\alpha\beta \times$ area OAE. Let OFD be the integral curve of OAB, that is, a curve such that any ordinate EF is proportional to the area OAE. Then ordinates similar to EF represent areas similar to OAE, and it follows that in order to transform the ordinates of OFD into velocity changes we have only to multiply by the value of $\alpha\beta$ as found in [29]. Likewise, if we multiply the ordinates of the acceleration curve OAB by α we shall have the corresponding value of the actual acceleration $\frac{dv}{dt}$.

From equation [22] we are now able to derive the values of y and to plot them if desired.

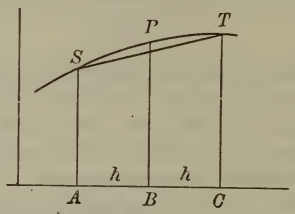


FIG. 7 DIAGRAM SHOWING METHOD OF FINDING TANGENT LINE TO A GIVEN CURVE

64 To derive the values of $\frac{dy}{dt}$ we proceed as follows: In Fig. 7 let SPT be an arc of any curve for which AS, BP and CT are three consecutive ordinates separated by equal spaces h. Then it is well known that for a second-degree parabola[1] passing through S, P and T the tangent at the point P is parallel to the chord ST. Now if the points S and T are not too far removed from P, a second-degree parabola will lie very close to the actual curve, and the tangent to the parabola at P will lie very close to the tangent to the given curve at the same point. That is, under the circumstances specified, it is clear that the two curves at P will have sensibly the same direction and hence the tangent found in this manner will give a good approximation to the true value. Hence, if we denote the ordinates by subscripts 1, 2, 3. we shall have

$$\left(\frac{dy}{dt}\right)_2 = \frac{y_3 - y_1}{2h} \qquad [33]$$

[1]Mathematical Treatment of Continuous Functions by Approximate Methods, Sibley Journal of Engineering, January 1894.

This gives a numerical method for finding $\frac{dy}{dt}$ for the curve of y on time at all except the first and last points.

65 For points such as S and T (Fig. 7) it has been shown[1] that the values of $\frac{dy}{dt}$ for the second-degree parabola are as follows:

$$\left(\frac{dy}{dt}\right)_1 = \frac{4y_2 - 3y_1 - y_3}{2h} \quad \text{[34]}$$

$$\left(\frac{dy}{dt}\right)_3 = \frac{y_1 + 3y_3 - 4y_2}{2h} \quad \text{[35]}$$

66 Instead of the preceding method of finding values of $\frac{dy}{dt}$ involving the numerical differentiation of the y curve, we may utilize [25] giving:

$$\frac{dy}{dt} = \left(\frac{a\alpha}{\beta}\right)\frac{dz}{dx} + 2\alpha bvz$$

or

$$\frac{dy}{dt} = \alpha\left(\frac{a\,dz}{\beta dx} + 2bvz\right) \quad \text{[36]}$$

The determination of $\frac{dy}{dt}$ in this manner involves the differentiation of the z or acceleration curve instead of the y curve, and may in certain cases be the more useful mode of procedure. In particular, it permits of finding the value of n for any point on the z curve in advance of the work required to determine the values of y.

67 This method permits therefore, in particular, the determination, independently of y, of the size of the chamber at the top and near the bottom and at one of two intermediate points as desired, thus indicating the general size and character of the chamber which a given acceleration curve will produce. To this end, from [23] we write

$$n = \frac{v' - v}{\alpha\left(\frac{a}{\beta}\frac{dz}{dx} + 2bvz\right)} = \left(\frac{R\beta}{v_2 - v_1}\right)\frac{v' - v}{\left(\frac{a}{\beta}\frac{dz}{dx} + 2bvz\right)} \quad \text{[37]}$$

68 At the initial instant when time = 0, we have

$$v' = v_3\frac{H - h_3}{H - h_1}$$

$$v = v_1$$

$$z = 0$$

[1]Mathematical Treatment of Continuous Functions by Approximate Methods, Sibley Journa of Engineering, January 1894.

Hence at the point for $T = 0$ we have

$$n_0 = \frac{v' - v_1}{\frac{a\alpha}{\beta}\frac{dz}{dx}} = \left(\frac{R\beta^2}{v_2 - v_1}\right)\frac{v' - v_1}{a\frac{dz}{dx}} \quad \text{[38]}$$

The value of $\frac{dz}{dx}$ is to be found as in [34] and then by substitution in [38] the value of n for the point where $y = 0$ is immediately found. A numerical example of these computations will be found in Appendix No. 3.

69 If the influence of governor control is neglected, $v' = v_3$ continuously and the equations become somewhat simplified as follows: For the value of n at any given point we have from [37]

$$n = \frac{v_3 - v}{\alpha\left(\frac{a}{\beta}\frac{dz}{dx} + 2bvz\right)} = \left(\frac{R\beta}{v_3 - v_1}\right)\frac{v_3 - v}{\frac{a}{\beta}\frac{dz}{dx} + 2bvz} \quad \text{[39]}$$

For the value of n at the beginning of the period when $z = 0$ and $v = v_1$ we have

$$n_0 = \frac{R\beta^2}{a\frac{dz}{dx}} \quad \text{[40]}$$

In equations such as [36], [37], [39] the algebraic sign of $\frac{dz}{dx}$ must be carefully noted.

70 It is seen that at the end of the period both $(v' - v)$ and $\frac{dy}{dt}$ become zero together, so that the value for this point becomes indeterminate. By the usual mathematical methods, the expression for n at this point may be evaluated and the resulting value of n found. In general, however, this will be quite unnecessary since the methods previously given provide for the determination of the size for any point as near the end as may be desired. Thus the determination of the size for the point where $x = 9$ on a scale of 10 will in general give a value sufficiently near for any possible practical purpose.

MAXIMUM VALUE OF y

Amount of Surge

71 The maximum value of y will be, of course, the value for the close of the period and when $v' = v = v_2$. Substitution in [22] gives

$$y = a\left(\frac{dv}{dt}\right) + b\ (v_2^2 - v_1^2)$$

72 The movement for ultimate steady conditions is $b\ (v_3^2 - v_1^2)$. The difference will be the surge. Hence we have

$$\text{Surge} = a\frac{dv}{dt} + b\,(v_2^2 - v_3^2) \quad \ldots\ldots\ldots\ldots\ldots\ldots [41]$$

or again substituting for $\frac{dv}{dt}$ we have

$$\text{Surge} = a\alpha z + b(v_2^2 - v_3^2) \quad \ldots\ldots\ldots\ldots\ldots\ldots [42]$$

The value of z used in [42] must, of course, be that for the close of the period or at the terminal end of the acceleration curve. The amount of surge may therefore be computed from [41] or [42], or inversely the characteristics of the acceleration curve may be adjusted in such manner as to secure a surge of any specified limiting value.

DETERMINATION OF SCALE RATIOS AND NUMERICAL RELATIONS FOR THE CASE OF NON-OSCILLATORY MOVEMENT

73 If the curve of acceleration for non-oscillatory movement is taken as *OABC* (Fig. 5), then the program of operation is exactly the same as that for any curve in general, and as already described. It should be especially noted, however, that in order to insure the conditions for a vanishing value of the surge, the relation between the values of z_{10} and $\left(\frac{dz}{dx}\right)_{10}$ must be adjusted so as to give a large value of β. This is apparent from [41] or [42]. By appropriate adjustment, therefore, the conditions may be determined so as to produce a surge of 0.5 ft., 0.1 ft., or any other value which may be considered negligible.

74 In case, however, the curve of acceleration is taken as *OABDE*, then at the point of coincidence with the axis of x we have $z = 0$ and $\frac{dz}{dx} = 0$ or otherwise $\frac{1}{\alpha}\frac{dv}{dt} = 0$ and $\frac{\beta}{\alpha}\frac{d^2v}{dt^2} = 0$ [28]. Hence, referring to [25], it appears that $\frac{dy}{dt}$ at this point will become zero independent of the values of β and α or in other words, for all values of β and α. It results that we are free in this case to choose β at will. The outline of the procedure is therefore as follows:

75 The area is found as in the previous case. As before

$$\alpha\beta = \frac{v_3 - v_1}{R}$$

Assuming β at will, gives

$$\alpha = \frac{v_3 - v_1}{R\beta}$$

We then proceed in exactly the same manner as before with the determination of the various quantities, resulting in the ultimate determination of the values of

n and of the size and form of the chamber. The value of y_{max} in this case is always

$$y_{max} = b\ (v_3^2 - v_1^2)$$

and equals the fall of water to the level for steady conditions for velocity = v_3. The value of the surge is of course zero by condition. The value of n for $T = 0$ is given by [38] as in the more general case. The value of n for the end of the period is indefinite as in the general case. This, however, is a matter of no significance since the dimensions and form of the chamber are sufficiently determined otherwise.

76 For varying values of β it is evident that since the area of the acceleration curve R remains the same, the value of $\alpha\beta$ will remain unchanged, and hence the entire history of the velocity in its change from v_1 to v_3. Also it is evident that α will vary inversely as β. Hence in the value of y, as given in [22], the term

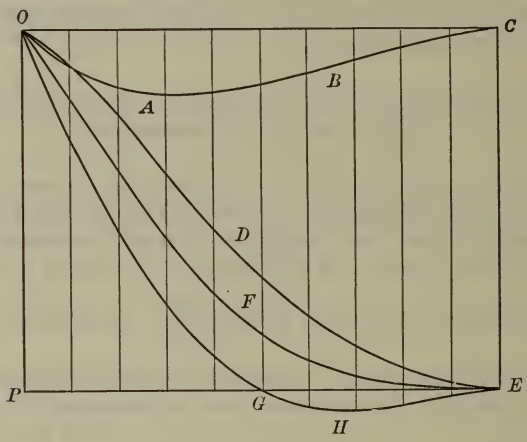


FIG. 8 DIAGRAM ILLUSTRATING METHOD OF FINDING MINIMUM CHAMBER FOR NON-OSCILLATORY MOVEMENT

$a\dfrac{dv}{dt}$ or $a\alpha z$ is the only one undergoing change, and this varies directly as α or inversely as β. Hence for example if β be doubled the values of $a\dfrac{dv}{dt}$ will be halved and the new values may be therefore immediately written from the old. The values of v' also remain the same and hence of $v' - v$. Hence for a determination of the form of chamber corresponding to a new value of β a part only of the entire program of computation is required, and a series of chambers corresponding to a series of values of β may therefore be quickly derived, once the results for a single case are in hand. At the initial level the area of the chamber will vary as β^2 as in the more general case and as shown by [38].

77 The value for n for an intermediate point is also readily found by the use of [37]. In this expression β is the only term which changes its value and hence for a given intermediate value of v we may find the value of $\dfrac{dz}{dx}$ once for all and then find a series of values of the ratio n. By the use of these direct methods values of n for the initial level and for two other points, one near the middle

and one near the end of the period, may be quickly found for a wide range of values of β. The values of y for such intermediate points are then found from [22] and such values of y and n together with those for the initial level will serve to determine the character of the chambers in general terms.

DETERMINATION OF MINIMUM CHAMBER FOR NON-OSCILLATORY MOVEMENT OF THE WATER

78 While it is true that with a form of acceleration curve such as *OABDE* the equations may be fulfilled for any value of β or any value of the time allowed, yet the case will fail physically if the time is too short. The conditions determining such limit may be investigated as follows: Rewriting equation [22] we have

$$y = a\frac{dv}{dt} + b\,(v^2 - v_1^2)$$

or

$$y = a\alpha z + b\,(v^2 - v_1^2)$$

79 Now plotting separately each of the two terms of the value of y and taking y positive downward, it is seen (Fig. 8) that they are of the forms indicated by *OABC* for $a\alpha z$ and *ODE* for $b\,(v^2 - v_1^2)$. In each case the curves have a horizontal tangent at the end of the period, and hence their sum, the curve of y will have a horizontal tangent at the same point, as is required for a maximum or minimum value.

80 Now change of β results in leaving the curve *ODE* the same, but in changing the scale of *OABC*. Hence with a large value of β and therefore a small value of α and a correspondingly small value of the ordinates of $a\alpha z = a\frac{dv}{dt}$ we shall have a curve of y somewhat as indicated by the line *OFE* in which, as shown, the water level sinks somewhat rapidly at first and then more and more slowly until it finally settles to E, reaching at this point its maximum value. Suppose, however, the value of β reduced to half its value for this case. Then the ordinates of *OABC* will be multiplied by two and adding such increased values to *ODE*, we shall have a result somewhat as in *OGHE*, where as seen, the curve cuts the level *PE* at G, drops below to a maximum at H and again rises through a point of inflection to a minimum at E. But

$$OP = b\,(v_3^2 - v_1^2)$$

Hence at the point G we have

$$y = a\frac{dv}{dt} + b\,(v^2 - v_1^2) = b\,(v_3^2 - v_1^2)$$

and since $\frac{dv}{dt} = \alpha z$, we find

$$a\alpha z = b\,(v_3^2 - v^2)$$

Substituting from [31] for α and solving for β

$$\beta = \frac{az\,(v_3 - v_1)}{bR\,(v_3^2 - v^2)} \qquad [43]$$

81 Now if we take the values of z and v corresponding to a point very near the end of the curve, say where $x = 9$ on a scale of 10, we shall find the value of β which would cause the y curve to dip across the line PE at this point. Some such value may be taken as the minimum value of β consistent with a substantial fulfillment of the condition of no surge.

82 The crossing of the y curve below PE and its later return upward while the velocity is still increasing toward its value v_3 represents an impossible set of physical conditions and the method of computation if carried forward would give an impossible form of chamber. Hence if we employ with the given acceleration curve a limiting value of β found as suggested in Par. 80, it will produce down to the level PE a form of chamber consistent with the physical conditions and which will drop the water level to PE at a time equal to 9β, for example. If the chamber is then continued down as conditions may require and of substantially the same area, it will mean simply a chamber in which the specified conditions of the acceleration curve at the end of the period would be imperfectly fulfilled, and in which there might result an infinitesimal surge. Such chamber would, however, practically fulfil all requirements for determining a

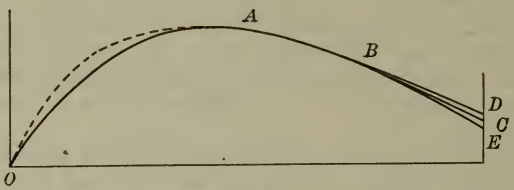


FIG. 9 DIAGRAM ILLUSTRATING RELATION BETWEEN FORM OF CHAMBER AND CHARACTERISTICS OF ACCELERATION CURVE

movement of the water substantially non-oscillatory, and might be taken as the smallest with which such conditions could be secured, at least on the basis of the assumed acceleration curve.

RELATION BETWEEN FORM OF ACCELERATION AND FORM AND DIMENSIONS OF CHAMBER

83 Suppose the assumed acceleration curve to be $OABC$, Fig. 9. Let the value of β be determined from [29]. Consider the result of slight modification at the end as indicated at BD or BE. For BD the value of z_{10} is increased and that of m, the tangent of the slope, is decreased. Hence β will be decreased. For BE the changes are in the reverse direction and the value of β will be increased. By a little trial adjustment the form of the curve may be so modified as to give within limits any desired value of β.

84 If now we suppose the form of chamber corresponding to the curve OAC to be determined, then it is evident that the areas $OABD$ and $OABE$ will differ but slightly from $OABC$. Hence to a first approximation R and $\alpha\beta$ in [30] and [31] will remain the same while α will vary inversely as β. It will result that the values of v and hence of b $(v^2 - v_1^2)$ will remain sensibly the same and hence that the change in the values of y, equation [22], will be determined by the change in the term $a \dfrac{dv}{dt} = a\alpha z$.

85 The new value of the surge, [41] or [42], is also quickly found as well as the values of n for the beginning and one or two other points on the curve by [37] and [38]. If these indications justify, the form and dimensions of the chamber may then be developed in detail by the regular treatment of the new curve *OABE* or *OABD*.

86 Referring further to the area at the level for $T = 0$ it appears from [38] that in such a case the value of n will vary directly as β^2 and hence for chambers of similar forms corresponding dimensions will vary directly as β. The resulting change in area or diameter of chamber at the level where $T = 0$ is thus apparent. If it is desired to hold the upper diameter or area within some specified limits, the slope of the curve may be correspondingly modified. Thus, if the change is in the direction *BE* the result will be an increase in the values of β and of the diameter of chamber where $T = 0$, both sensibly in the same ratio. If at the same time, however, the slope of the curve at the origin is increased, as indicated by the dotted line, this latter modification will serve to counteract more or less the change due to varying β and hence by adjustment between the two any specified or limiting value of this diameter may be determined.

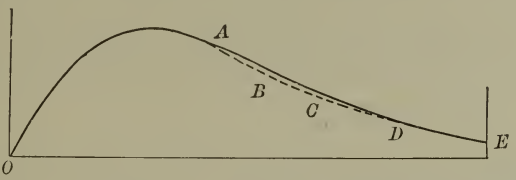


FIG. 10 DIAGRAM ILLUSTRATING RELATION BETWEEN FORM OF CHAMBER AND CHARACTERISTICS OF ACCELERATION CURVE

87 By the aid of [30], [31], [32], [37] the general results of any small modification in the form of the acceleration curve may be analyzed, showing the character of change in the form of the chamber. Thus, in Fig. 10, let the original curve indicated by the full line be modified as indicated by the dotted line. For the purposes of such an analysis the new curve may be divided into the following parts:

OA and *DE* remaining the same as before
AB with increased negative slope and decreased ordinate
BC with the same slope sensibly and decreased ordinate
CD with decreased negative slope and decreased ordinate.

88 The analysis need not be given in detail. Having in view the direction and relative quantitative amounts of change in the various quantities involved, the changes in the area of chamber will be as follows:

Period Corresponding to	*Change in Area of Chamber*
OA	decrease
AB	from decrease to small and uncertain and then increase
BC	increase
CD	from increase to small and uncertain
DE	small and uncertain

89 There are many other minor points of relationship between the form of the curve of acceleration and the resulting form of the chamber, the amount of surge, the characteristics of the movement of the water, etc., which will occur to anyone dealing with actual problems, but which are scarcely of sufficient importance to warrant further detailed notice. It results, however, that the acceleration curve may by such means be quickly and readily adjusted to develop within limits a chamber of any desired general character of form, and to fulfil any predetermined or specifiedprogram as regards the movement of the water surface, extent of surge and time required.

APPENDIX No. 3

ILLUSTRATIVE CASES

90 Two or three illustrative cases may be drawn from the problem which has given rise to the development of this particular method. This problem is concerned with the control of the water in the Los Angeles aqueduct at one of the power sites where the principal conditions are as follows:

91 The water is to be led through a cement lined rock tunnel a distance of about 41,000 ft. to the power drop whence a penstock of about 3150 ft. in length will lead to the power house. The upper portal of the tunnel is under a positive head varying with the stage of the reservoir but which may be taken at 85 ft. maximum. The total drop through the tunnels is 58.8 ft., made up of two gradients, 26,800 ft. for the first section on a gradient of 0.001 and 14,100 ft. for the second section on a proposed gradient of 0.0023. The fall at the power drop is 808 ft. and the total gross head is 808 + 58.8 + 85 = 951.8 or 952 ft.

92 The maximum amount of water to be handled is 1000 cu. ft. per sec. and the actual amount will be in accordance with the momentary power demand

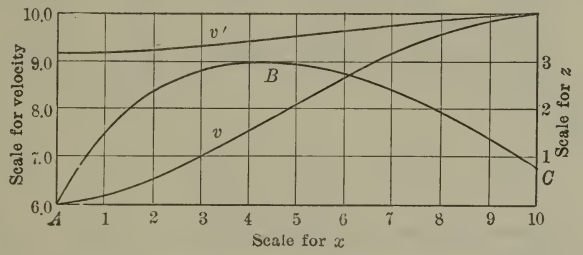


FIG. 11 DIAGRAM SHOWING TYPE CURVE OF ACCELERATION WITH RESULTING VELOCITIES PLOTTED ON TIME

up to this value as a maximum. The two sections of the tunnel are not of the same size, and the velocities of flow will therefore not be equal. For purposes of preliminary study, however, a uniform conduit of 100 sq. ft. section was assumed, thus giving a maximum velocity of 10 ft. per sec. It thus appears that under maximum conditions the tunnel as a whole will contain some 128,125 tons of water, which moving at a velocity of 10 ft. per sec. would have a kinetic energy of 200,000 foot-tons in round numbers. Again, it appears from computations which need not here be repeated that without the aid of a surge chamber in developing an acceleration head the time of acceleration from a flow of 600 cu. ft. per sec. to 990 cu. ft. per sec. when 1000 ft. are called for would be some 40 to 50 seconds. To relieve the conditions which would result from the acceleration or retardation in this manner of the 7.76 miles of water column during periods of change for both demanded and rejected load, a surge chamber of cement lined rock excavation, preferably of conical or vase-like form, is contemplated at the end of the lower tunnel and at the top of the power drop.

93 Many alternative studies have been made for this chamber based on a variety of operative conditions. Certain of the preliminary studies based on an assumed uniform section of tunnel may be here used for illustrative purposes.

CASE 1

94 For the numerical coefficients $a = \frac{L}{g}$ and b, we have in the present case

$$a = 1285$$
$$b = 1.2$$

also

$$v_1 = 6.0$$
$$v_3 = 10.0$$

For the friction head

$$h_f = \left(1.2 - \frac{1}{2g}\right)v^2 + 20 = 1.1844\,v^2 + 20$$

also

$$H = 950$$

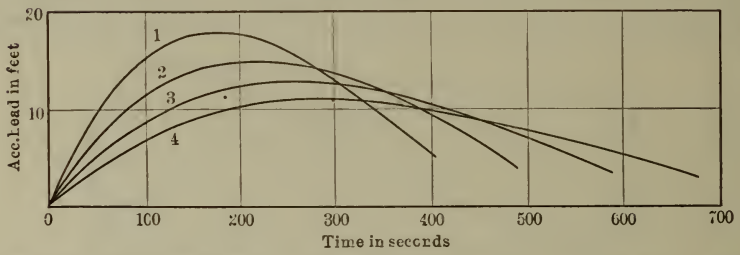


FIG. 12 DIAGRAM SHOWING ACCELERATION PLOTTED ON TIME FOR FOUR DIFFERENT CASES

(1) Acc. head for $\beta = 40.70$
(2) Acc. head for $\beta = 48.95$
(3) Acc. head for $\beta = 58.80$
(4) Acc. head for $\beta = 68.80$

95 For the type curve of acceleration the curve ABC (Fig. 11), was assumed. The numerical values of the ordinates, which we may denote by z, are given in column 2 of Table 1, the scale being entirely arbitrary and these numbers at present having no physical meaning. The assumed curve is first integrated by any of the well known methods and the resulting values placed in column 3. Trapezoidal integration has been used throughout in this preliminary work as will appear from an examination of the values. The total area is 21.415. This is the R of the formulae. We next find $m = \frac{dz}{dx}$ at the end of the curve by the use of [35]

$$1.93 \times 1 = 1.93$$
$$1.39 \times 4 = 5.56$$
$$0.75 \times 3 = 2.25$$

Then

$$1.93 + 2.25 = 4.18$$

and

$$4.18 - 5.56 = -1.38$$

The interval $h = 1$. Hence

$$m = -\frac{1.38}{2} = -0.69$$

TABLE 1 SHOWING TYPICAL FORM OF COMPUTATION

1	2	3	4	5	6	7	8	9
x	z		v	$v^2-v_1^2$	$b(v^2-v_1^2)$	$a\frac{dv}{dt}$	y	$2\frac{dy}{dz}$
0	0	0	6.000	0	0	0	0	18.100
1	1.47	0.735	6.139	1.688	2.026	7.320	9.346	19.284
2	2.36	2.650	6.502	6.276	7.531	11.753	19.284	20.263
3	2.83	5.245	6.995	12.930	15.516	14.093	29.609	20.804
4	3.00	8.160	7.547	20.957	25.148	14.940	40.088	20.828
5	2.95	11.135	8.111	29.788	35.746	14.691	50.437	19.953
6	2.73	13.975	8.649	38.705	46.446	13.595	60.041	18.331
7	2.38	16.530	9.134	47.430	56.916	11.852	68.768	15.630
8	1.93	18.685	9.542	55.050	66.060	9.611	75.671	11.560
9	1.39	20.345	9.858	61.180	73.416	6.912	80.328	6.404
10	0.75	21.415	10.060	65.200	78.240	3.730	81.970	0

	10	11	12	13	14	15	16
x	h	$H-h$	v	$v'-v$	n	D	Y
0	62.64	887.36	9.145	3.145	17.02	45.39	43.20
1	71.96	878.04	9.242	3.103	15.76	43.67	52.55
2	81.82	868.18	9.348	2.846	13.75	40.79	62.48
3	92.05	857.95	9.460	2.465	11.60	37.47	72.81
4	102.40	847.60	9.575	2.028	9.53	33.96	83.29
5	112.61	837.39	9.691	1.580	7.75	30.62	93.64
6	122.07	827.93	9.802	1.158	6.18	27.35	103.24
7	130.67	819.33	9.905	0.771	4.83	24.18	111.97
8	137.45	812.55	9.988	0.446	3.78	21.38	118.87
9	142.01	807.99	10.044	0.186	2.84	18.54	123.53
10	143.59	806.41	10.060	0	.....	.....	125.18

96 We then find by trial, using the equations given in Appendix No. 2, the values $v_2 = 10.06$, $\beta = 48.95$ and $a = 0.003873$. Hence, the total time for the period = 489.5 seconds = 8.16 minutes. The time required on the basis of the assumed curve is thus determined. We have, therefore,

$$a\beta = \frac{v_2 - v_1}{R} = 0.1896$$

We next find by inserting the proper numerical values in [42]

$$\text{Surge} = 5.17$$

97 Taking the factor $a\beta = 0.1896$ as explained in Appendix No. 2 and using it as a multiplier, we convert column 3 of Table 1 into velocity change, and

adding $v_1 = 6$ we have the velocity history on time as given in column 4. We then find values of $(v^2 - v_1^2)$ as in column 5 and multiplying by $b = 1.2$ we have values of $b\ (v^2 - v_1^2)$ as in column 6.

98 From the value a (Par. 96)

$$aa = 1285 \times 0.003873 = 4.98$$

Using this as a factor, we convert column 2 into values of $a \frac{dv}{dt}$ as in column 7. We then add the values in columns 6 and 7 finding values of y and enter in column 8.

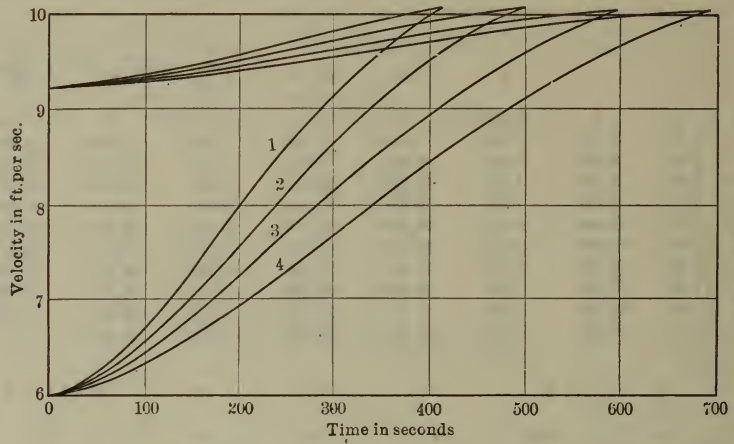


FIG. 13 DIAGRAM SHOWING VELOCITIES PLOTTED ON TIME FOR FOUR CASES AS IN FIG. 12

(1) Velocities v and v' for $\beta = 40.70$
(2) Velocities v and v' for $\beta = 48.95$
(3) Velocities v and v' for $\beta = 58.80$
(4) Velocities v and v' for $\beta = 68.80$

99 Next by the use of [33], values of $2 \frac{dy}{dx}$ for the various points 0 - - - - 9, inclusive, are obtained and entered in column 9. Evidently

$$\frac{dy}{dt} = \frac{2 \frac{dy}{dx}}{2\beta}$$

This division by 2β may be taken as a multiplication in the final operation for n as noted below.

100 We next find values of h by adding together values of the acceleration head, column 7, and of $h_f = 1.1844\ v^2 + 20$. These results are entered in column 10 and subtracting them from the gross head, taken here as 950, we have values of $(H - h)$ as in column 11. We then find values of the virtual velocity v' by equation [14], Appendix No. 2, and enter the results in column 12. Subtracting v (column 4) from these, we find values of $(v' - v)$ as in column 13. Dividing the values in column 13 by those in column 9 and multiplying by

$2\beta = 97.9$ we find values of n as in column 14. These converted into diameter of chamber give the values in column 15.

101 The upper diameter of 45.4 ft., it will be remembered, is for the water level corresponding to a steady velocity of 6 ft. per sec. This will be 43.2 ft. below static level. Hence, adding 43.2 to the values of y in column 8 we find values of Y the distance of the water surface below static level. These are given in column 16.

102 The principal results are shown graphically in Figs. 12, 13 and 14. The dotted line shows the surge chamber slope continued up to the static level

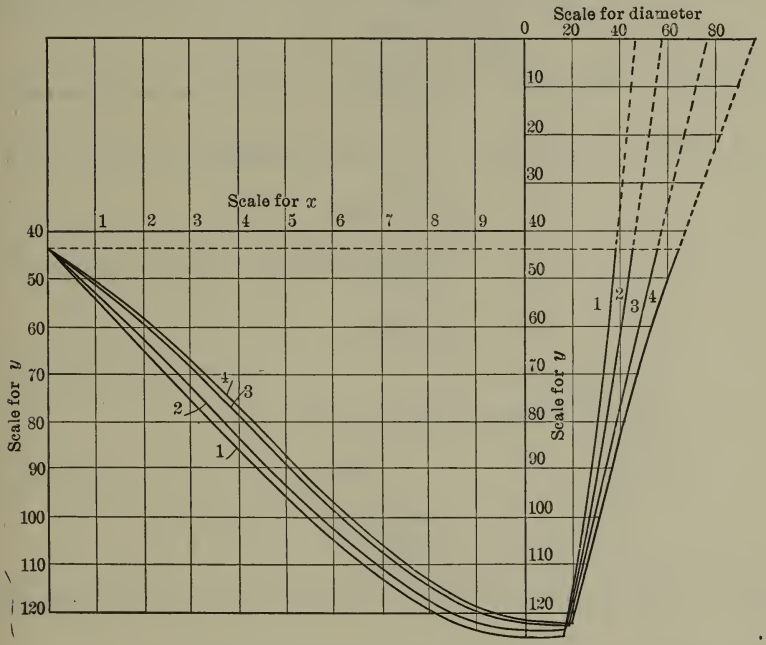


Fig. 14 Diagram showing Values of y plotted on Time and Profiles of Surge Chambers for Four Cases as in Fig. 12

Curves 1 refer to case for $\beta = 40.70$
Curves 2 refer to case for $\beta = 48.95$
Curves 3 refer to case for $\beta = 58.80$
Curves 4 refer to case for $\beta = 68.80$

and indicates, therefore, the general dimensions as some 60 ft. in diameter at the top and 18 ft. at the bottom. Such a chamber then with its given dimensions between the level 43.2 and the bottom would determine for a change in velocity from 6 ft. per sec. to 10.06 ft. per sec., an acceleration and velocity history as indicated, and a maximum fall of water ot 81.97 ft. below the level for 6 ft. per sec. steady conditions. This is 125.17 ft. below static level and 5.17 ft. below the level for 10 ft. per sec. steady flow. The surge is, therefore, 5.17 ft. as previously determined.

103 We may next illustrate the determination of the value of n for $x = 0$, $x = 3$ and $x = 9$ by the use of [37] and [38]. For $\left(\frac{dz}{dx}\right)_0$ we use [34], giving

$$\frac{dz}{dx} = 1.76$$

Then from [38] we have

$$n_0 = \frac{21.415 \times 2396.6 \times 3.145}{4.06 \times 1285 \times 1.76} = 17.35$$

and $D = 46.0$. Likewise for $x = 3$ we find $\frac{dz}{dx} = 0.32$ and substituting the values in [37] we find $n = 11.33$ and $D = 37$. For $x = 9$ we find

$$\frac{dz}{dx} = -0.59$$

$$n = 2.75$$

and

$$D = 18.3$$

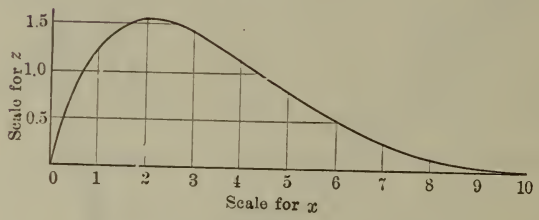


FIG. 15 DIAGRAM SHOWING TYPE CURVE FOR NON-OSCILLATORY MOVEMENT

104 These various results agree as well with those of Table 1 as could be expected considering the open spacing of ordinates and trapezoidal integration. With closer spacing of ordinates and the use of parabolic rules for integration the difference between the two sets of results will become a vanishing quantity.

105 Suppose next the end of the acceleration curve raised, giving to z values as follows:

$$z_8 = 1.95$$
$$z_9 = 1.45$$
$$z_0 = 0.85$$

We then find

$$\left(\frac{dz}{dx}\right)_{10} = -0.65$$

and

$$\beta = 40.7 \text{ seconds}$$

106 We may assume that the values of R and of v_2 for the transformed curve will differ only slightly from the original values. This assumption will evidently involve no error of significant amount. We may then find immediately the surge from [41] or [42], thus

$$\text{Surge } 1285 \times 0.004658 \times 0.85 + 1.2 \times 1.2 = 6.53$$

107 In the same manner as for the original curve we find by [37] and [38] values of n for $x = 0$, $x = 3$, $x = 9$. For $x = 0$ it is evident from [38] that n varies as β^2. Hence

$$n_0 = 17.35 \times \left(\frac{40.7}{48.95}\right)^2 = 12.03$$

$$D = 38.2$$

For $x = 3$ we find by substitution in [37]

$$n = 9.14$$
$$D = 33.3 \text{ ft.}$$

For $x = 9$ we have $\frac{dz}{dx} = -0.55$ and we find

$$n = 2.36$$
$$D = 16.9 \text{ ft.}$$

The new value of α will be:

$$\alpha = \frac{0.003873 \times 48.95}{40.7} = 0.004658$$

This will give for the new values of y

$$y_3 = 32.5$$
$$y_9 = 81.7$$

These results give an outline of the size and form of the resulting chamber in Fig. 14.

TABLE 2 PRINCIPAL RESULTS FOR FOUR CASES DERIVED FROM THE SAME TYPE ACCELERATION CURVE

	z_8	z_9	z_{10}	β	Time, Min.	Surge	D_0	D_3	D_9
Original Curve	1.93	1 39	0.75	48.95	8.16	5.17	46.0	37.0	18.3
Change 1	1.95	1.45	0.85	40.70	6.78	6.53	38.2	33.3	16.9
Change 2	1.91	1.35	0.67	58.80	9.80	4.21	55.4	41.2	19.5
Change 3	1.89	1.31	0.60	68.80	11.47	3.56	64.8	44.8	20.7

108 If the form of the acceleration curve near the end be slightly modified in the opposite direction, there will result for each change a new case, giving larger sized chambers and larger values of the time. The results of such changes may be determined in the same manner as outlined in Par. 106 and Par. 107. The principal features for the original and for three changes in the acceleration curve are given in Table 2, and are shown graphically in Figs. 12, 13 and 14.

109 These results for the three changes are of course only approximations, since the change at the end will affect the total area slightly and hence the entire velocity history. The more exact values may of course be found by going

TABLE 3 VALUES OF ORDINATES OF CURVE IN FIG. 15

x	z	x	z
0	0	5.5	0.55
0.5	0.80	6.0	0.42
1.0	1.22	6.5	0.30
1.5	1.44	7.0	0.22
2.0	1.50	7.5	0.16
2.5	1.44	8.0	0.12
3.0	1.30	8.5	0.08
3.5	1.14	9.0	0.04
4.0	0.98	9.5	0.02
4.5	0.83	10 0	0.
5.0	0.68	....	....

through the case in the regular manner. Assuming the change in the acceleration curve small, however, the errors will be small, and the values found in

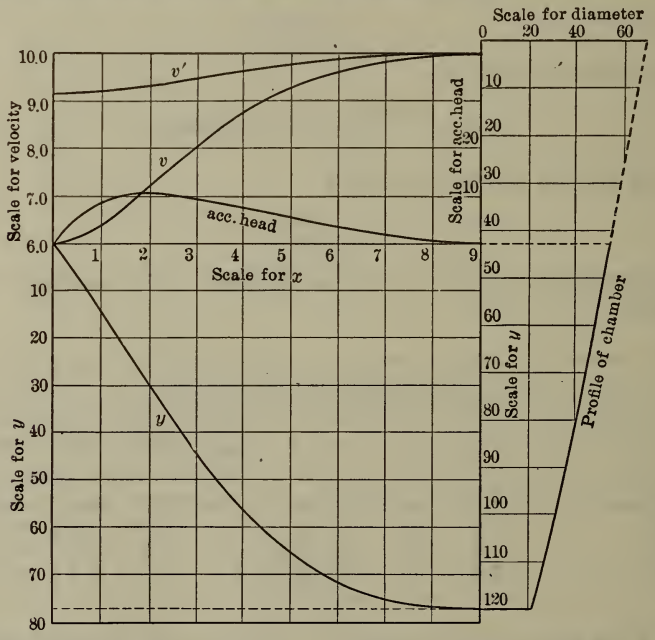


FIG. 16 DIAGRAM SHOWING GENERAL RESULTS FOR THE ACCELERATION CURVE OF FIG. 15

Table 2 will give a closely approximate idea of the general results as to time, extent of surge and size and form of chamber.

Case 2

110 As an illustration of a case of practically non-oscillatory movement reference may be made to the acceleration curve of Fig. 15 where the portion employed is that lying between $x = 0$ and $x = 9$. The value of the ordinates of this curve are given in Table 3.

111 The general method is exactly the same as for the preceding cases and need not be developed in detail. The principal results with the form and dimensions of the chamber and the graphical history of certain of the quantities are shown in Fig. 16.

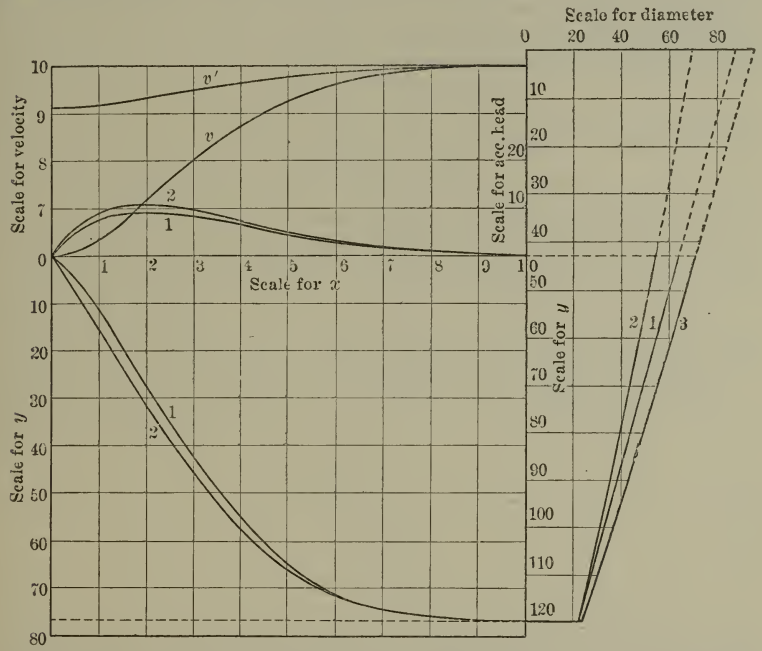


Fig. 17 Diagram showing General Results for Non-Oscillatory Movement with Acceleration Curves as shown

Curves 1 refer to case for $\beta = 120$
Curves 2 refer to case for $\beta = 108$
Curves 3 refer to case for $\beta = 120$ and neglecting governor control

Case 3

112 If the entire extent of the curve in Fig. 15 be used, the value of β becomes indefinite as we have seen. If we assume $\beta = 120$ seconds, we may then work through the case as before. The results for such a case are shown graphically in Fig. 17. The time is 20 minutes and the surge is zero by condition.

113 In order to determine the minimum time and the minimum size of the chamber with which the surge, with this form of acceleration curve, may still be kept near the vanishing point, we substitute in [43] the values for the curve at $x = 9$ and find $\beta = 108$. This gives a time of 972 seconds or 16.2 minutes

within which the water level will descend 76.8 ft. while the velocity will rise from 6 ft. per sec. to 9.988 ft. per sec. This is a substantial fulfillment of the conditions as required. If the velocity goes on to 10 ft. per sec. the water level will fall below the level for steady motion with $v = 10$, and an infinitesimal surge will result. The form and dimensions of the chamber corresponding to this condition are shown in Fig. 17.

Case 4

114 This case is the same as Case 3 with $\beta = 120$ except for the omission of the governor control. The rate of penstock flow is taken as the equivalent of a velocity of 10 ft. per sec. in the main conduit throughout the entire period. The simplification in the numerical work will be readily apparent without detailed description. The results are shown in Fig. 17.

APPENDIX No. 4

THE LAW OF KINEMATIC SIMILITUDE APPLIED TO THE SURGE CHAMBER

115 Since the preparation and publication of the preceding portion of the paper, the application of the principles of kinematic similitude to the surge chamber equations has been worked out, thus making it possible to extend the results determined for any one case to a series of other cases related to the original, or "type" case, by certain determinate ratios.

116 The method is so simple and promises so ready a means thus of utilizing, for a series of cases, the results of any one computation, that it is here appended as a supplement to the original paper.

117 It must first be noted that the general case with full governor control and constant power during the period of change does not adapt itself to such mode of treatment, except under the restriction of a constant relation between the total head and the friction head, a condition which would seriously narrow the field of possible use.

118 In the following discussion, therefore, we shall assume the case of constant volume of flow at the nozzles during the period of change, corresponding in the general equations of the paper, to the identity of v_2 and v_3.

119 With this understanding, we may write the fundamental equations as follows (see [1] and [2]):

$$y = \frac{L}{g}\frac{dv}{dt} + b\,(v^2 - v_1^2) \qquad [44]$$

$$n = \frac{v_2 - v}{\frac{dy}{dt}} \qquad [45]$$

120 Now it is assumed that we are to find, for a series of different cases, the conditions for a corresponding series of surge chambers, in all of which the acceleration curves will be similar. In other words, the type acceleration curve is to be the same throughout the series, although the actual acceleration, the acceleration head, the actual time and the vertical water movement may vary indefinitely from one case to another, according to the value of the units applying in any given case.

121 Consider any two acceleration curves related in this manner as indicated in Fig. 18. Then, to every point on the one which we take as the type curve, there will correspond a point on the other; such, for example, as the points P and Q.

122 Let the various characteristics of the two cases be related as follows:

Ratio of conduit lengths, or L ratio $= p$

Ratio of coefficients for friction and acceleration head, or b ratio $= q$

Ratio of velocities, or v ratio $= r$

Ratio of times, or t ratio $= s$

Then we have, immediately, as follows:

$$v^2 \text{ ratio} = r^2$$

$$b(v^2 - v_1^2) \text{ ratio} = qr^2$$

But all terms of [44] must be affected by the same ratio; hence, we must have

$$y \text{ ratio} = qr^2$$

$$\frac{L}{g}\frac{dv}{dt} \text{ ratio} = qr^2$$

$$\frac{dv}{dt} \text{ ratio} = \frac{qr^2}{\ }$$

But the $\frac{dv}{dt}$ ratio must equal the quotient of the v and t ratios. Hence

$$\frac{dv}{dt} \text{ ratio} = \frac{r}{s}$$

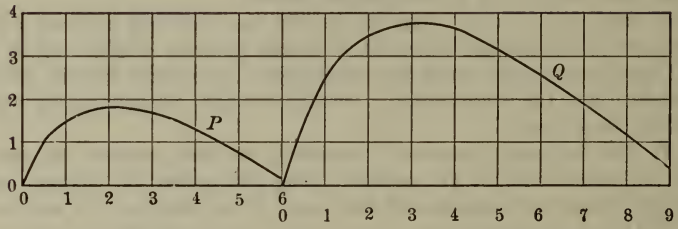


FIG. 18 CORRESPONDING CURVES OF ACCELERATION

Hence we have

$$\frac{qr^2}{p} = \frac{r}{s}$$

or

$$s = \frac{p}{qr}$$

123 This gives the value of the time ratio in the two cases. Again, the $\frac{dy}{dt}$ ratio must equal the quotient of the y and t ratios; or

$$\frac{dy}{dt} \text{ ratio} = \frac{qr^2}{s} = \frac{q^2r^3}{p}$$

and

$$n \text{ ratio} = v \text{ ratio} \div \frac{dy}{dt} \text{ ratio}$$

or

$$n \text{ ratio} = \frac{pr}{q^2r^3} = \frac{p}{q^2r^2}$$

124 Let $d_o A_o D_o F_o$ denote, respectively, the diameter of the conduit (assumed circular), area of conduit, diameter of surge chamber and area of the surge chamber, all for the type case, with similar notation without subscripts for the general case.

125 Then

$$n_o = F_o \div A_o$$
$$n = F \div A$$

and

$$n \text{ ratio} = \frac{p}{q^2 r^2} = \frac{F}{F_o} \frac{A_o}{A}$$

Hence

$$\frac{F}{F_o} = \frac{p}{q^2 r^2} \frac{A}{A_o} = \frac{p}{q^2 r^2} \left(\frac{d}{d_o}\right)^2$$

Whence

$$\frac{D}{D_o} = \frac{\sqrt{p}}{qr} \left(\frac{d}{d_o}\right), \text{ or } \frac{\sqrt{p}}{qr} \sqrt{\frac{A}{A_o}}$$

and

$$\text{volume ratio} = \frac{Fy}{F_o y_o} = \frac{p}{q} \left(\frac{d}{d_o}\right)^2, \text{ or } \frac{p}{q} \frac{A}{A_o}$$

126 Collecting, we have ratios as follows:

a The ratio of all similar vertical dimensions, such as maximum movement of water level, movement for final steady conditions, movement for corresponding parts of the acceleration curve, etc., will equal qr^2

b The ratio of the time intervals for the whole, or for corresponding parts of the transition phenomena, will equal $\frac{p}{qr}$

c The ratios of the diameters of the chamber at corresponding levels will be

$$\frac{\sqrt{p}}{qr} \left(\frac{d}{d_o}\right), \text{ or } \frac{\sqrt{p}}{qr} \sqrt{\frac{A}{A_o}}$$

d The ratios of the volumes swept through by the water surface in the two cases will be

$$\frac{p}{q} \left(\frac{d}{d_o}\right)^2, \text{ or } \frac{p}{q} \frac{A}{A_o}$$

127 The application of these principles will be readily seen from an illustrative example.

128 Suppose, in a given case, we have

$$L^\circ = 6000 \text{ ft.}$$
$$b_o = 0.16$$
$$d_o = 4 \text{ ft.}$$

and that a certain chamber has been worked out for 30 ft. diameter at level for $v = 4$ ft-sec., and 24 ft. in diameter at level for $v = 7$ ft-sec., and that for demanded load with change in velocity from 4 to 7 ft-sec., a time of 4 minutes and a maximum drop of water level of 6.28 ft. is produced with a drop of water for steady conditions of 5.28 ft., or a surge (as defined in the paper) of 1 ft.

129 From these results for a case where

$$L = 10{,}000 \text{ ft.}$$
$$b = 0.21$$
$$d = 6 \text{ ft.}$$

and

lower velocity, $v_1 = 5$ ft-sec.

we have

$$L \text{ ratio} = p = 10 \div 6 = 1.66$$
$$b \text{ ratio} = q = 21 \div 16 = 1.3125$$
$$v \text{ ratio} = r = 5 \div 4 = 1.25$$

Then upper velocity $v_2 = 1.25 \times 7 = 8.75$, vertical movement ratio $= qr^2 = 2.051$, and the maximum drop of water level will be 12.88 ft., the movement to steady conditions will be 10.83 and the surge will be 2.05 ft. The time ratio is $p \div qr = 1.66 \div (1.3125 \times 1.25) = 1.016$. Hence the corresponding time will be 4.06 minutes, and the diameter ratio is $\frac{\sqrt{p}}{qr} \times 1.5 = 1.182$. Hence the chamber will be 35.5 ft. diameter at top, and 28.4 ft. at bottom, and the volume will be 2.84 times the original.

130 These equations and principles suggest immediately the possibility of the use of models of surge chambers and the extension of the results of observations on any model chamber to a corresponding full-sized case. They show, also, how to select a convenient size of model and possible operating conditions in such manner as to correspond to any stated actual case. Thus, for example, take the case used for illustration in the paper:

$$L = 41{,}320$$
$$b = 1.2$$
$$v_1 = 6$$
$$v_2 = 10$$
$$A = 100 \text{ sq. ft.}$$

131 Now suppose, in the model, we have

$$L = 12$$
$$b = 0.10$$
$$v_1 = 3$$
$$v_2 = 5$$
$$A = 0.005 \text{ sq. ft.}$$

Then

$$p = 3443$$
$$q = 12$$
$$r = 2$$

Vertical distance ratio $= qr^2 = 48$

Time ratio $= p \div qr = 143.4$

Diameter ratio $= (\sqrt{p} \div qr)\ \sqrt{20{,}000} = 345.6$

Volume ratio $= 5{,}740{,}000$

These ratios correspond to an entirely workable size of model.

132 Thus, suppose, with a tapering model chamber, the water level for an increase in v from 3 to 5 ft-sec. descends h feet in t seconds, and from a diameter of 3 in. to one of 1.5 in. Then in the actual case for a corresponding increase in v from 6 ft-sec. to 10 ft-sec. and along the same type of acceleration curve, there will be required at the upper water level a diameter of 86.4 ft. and at the lower level a diameter of 43.2 ft., and the water level will drop $48h$ ft. in $143.4t$ seconds.

DISCUSSION

W. R. Eckart, Jr. The timeliness and practical value of Professor Durand's paper can hardly be overestimated especially in connection with the hydroelectric development being carried on on the Pacific Coast. Numerous cases might be cited, where in the construction of these plants the addition of a properly designed surge chamber would help materially in the regulation of the plant.

As shown by Professor Durand, the methods generally used in the solution of these problems, depending as they do upon modifications of the general theoretical formulae, or the adoption of empirical ones based upon scattered results of actual practice, lead at the best only to approximate results and then only for a chamber of uniform section.

In 1908 in connection with the installation of an additional unit at the power house of the Snow Mountain Water and Power Company, the problem of a surge chamber was taken up. A contour survey of the ground at the proposed location was made and a chamber of irregular shape, adapted to the contours of the site, was tentatively chosen. Although the walls of this chamber had a slope of about 2 to 1, it was necessary for the purposes of calculation to adopt an equivalent chamber of uniform area of surface. In consequence the influence of the tapering section upon the surge did not appear in the derived curves showing the elevation of the water surface at different time intervals after changing the flow in the conduit. The results showed that a chamber of somewhat reduced capacity could be used and calculations were made for a second preliminary scheme using a smaller cylindrical reservoir, the capacity of which could be supplemented later by the addition of one or more similar reservoirs, as the enlargement of the plant might demand. Later through an inference from an article[1] descriptive of the Urft dam and hydroelectric plant, it was thought that a conical form of chamber would be more desirable than a cylindrical one, and a sketch diagram for the same was made.

In the calculations by any method, the accuracy of the determination of the drop in the surge chamber will depend very largely upon the values assumed for the coefficient of friction for the particular conduit under consideration. In the Snow

[1] London Engineering, Aug. 21, 1908.

Mountain proposition, owing to the varying section and character of the conduit, it was deemed advisable to determine the actual value of these coefficients by experiment. From the tunnel intake to the point of connection to the proposed surge chamber the conduit has a total length of 7097 ft. made up of the following different sections:

An open cut to the tunnel portal, the latter being covered with a grizzly.
5658 ft. of timber lined tunnel, with concrete floor, the sectional area being 38.1 sq. ft.
218 ft. of riveted steel pipe, sectional area 23.76 sq. ft.
384 ft. of wood stave pipe, sectional area 38.5 sq. ft.
837 ft. of riveted steel pipe, sectional area 23.76 sq. ft.

For the purposes of the experiment, a 1-in. pipe was tapped into the main pipe at the point at which it was proposed to make connection with the surge reservoir; to this pipe a series of manometer tubes were connected, so arranged that the hydraulic head could be determined through a range of 30 ft. Simultaneous readings were made at these gages, and of the water surface at the main reservoir, the water being measured at the power house, using a current meter in the concrete tail race of uniform section; several weir measurements were made of the discharge, where the same was less than 85 seconds per ft., serving as a check on the gagings. Incidental to the determination of the loss of head at the surge reservoir site, a number of observations were made of the water levels in the several shafts of the tunnel, giving the loss of head due to entrance through the grizzly and portal, the loss of head in the tunnel proper, the loss of head for the steel and wood pipes combined, and from a carefully calibrated standard gage, the total loss of head for the entire conduit to the power house. From these data the coefficients for the conduit were determined and for convenience in calculating, a conduit of uniform section of equivalent diameter was assumed.

At the time this work was being carried on, Professor Durand was engaged on a somewhat similar problem in connection with the Los Angeles aqueduct, which resulted in the method of solution as developed in this paper. The methods therein described have since been applied in the treatment of the Snow Mountain proposition, and also to a study of a similar problem in connection with a tunnel project for the Colgate plant of the Pacific Gas & Electric Company. Any one who has attempted the solution of the surge chamber problem by the methods previously

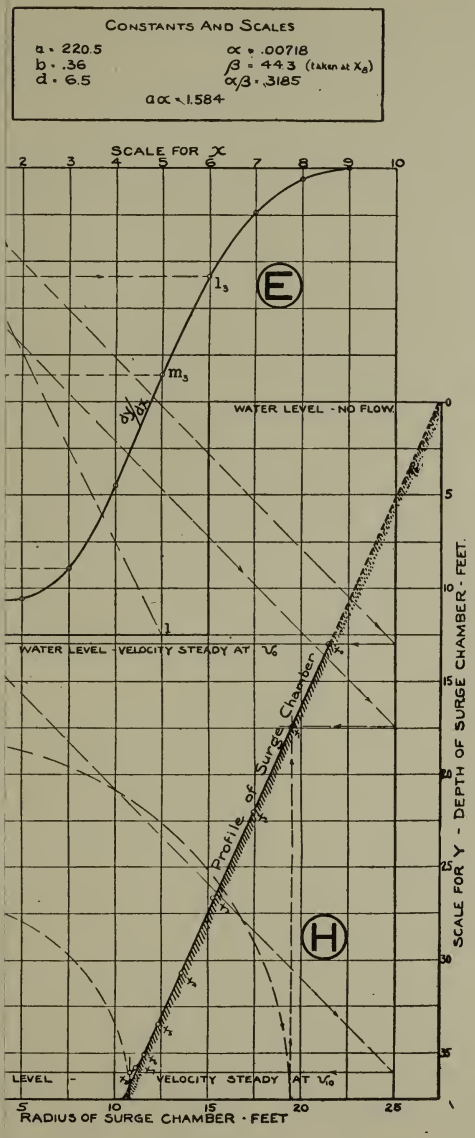
available will appreciate the great advantage of this new solution offered by Professor Durand in that the size and form of the chamber are definitely and directly determined for a given set of conditions which are chosen to represent the most severe demands liable to be made on the surge reservoir. The selection of a form of acceleration curve on which to base the calculations presents no difficulty after one or two trials.

In working out the Snow Mountain problem, application of graphical methods has been made. This, while not being perhaps as rapid as the direct computation, permits of an equal degree of accuracy in the results, at the same time giving directly the different curves involved and any error becomes immediately apparent from an inspection of the curves thus developed. The graphical solution, involving as it does graphic integration, differentiation, multiplication, division, involution and evolution, addition and subtraction, may prove of interest to those who prefer such methods. If we neglect the effect of governor control, it will be found, as shown in Fig. 17 of the paper, that the resulting chamber is a trifle larger in size, and therefore allows a certain factor of safety in the solution of the problem, which may be considered desirable from the uncertainty of the coefficients used in determining the friction in the conduit. The graphical solution given is worked out on this basis.

In Fig. 19, Diagram A, are plotted the assumed form of acceleration curve, the scales for the ordinates and abscissae of which are entirely arbitrary. However, if we should multiply the ordinates z by a, the scale for acceleration, and the abscissae by β, the scale for time, we should then have an acceleration curve plotted to a time base, but as this would be of the same general form as that chosen in Diagram A, Fig. 19, the solution of our problem would not have been advanced thereby.

Starting with Diagram A, Fig. 19, the first step is to integrate the acceleration curve, which may be done graphically as follows: To compare the graphical results with the numerical ones, the processes for which are outlined in the paper, trapezoidal integration has also been used here.

Naturally, with this method the degree of accuracy depends upon the number of ordinates taken, especially when dealing with the first part of the acceleration curve, or the same may be done by the use of graphical methods, which will obtain with


CONSTANTS AND SCALES
a = 220.5
b = .36
d = 6.5
α = .00718
β = 443 (taken at X8)
α/β = .3185
aα = 1.584
SCALE FOR X
2 3 4 5 6 7 8 9 10
E
H
WATER LEVEL - NO FLOW
WATER LEVEL - VELOCITY STEADY AT V6
LEVEL - VELOCITY STEADY AT V10
Profile of Surge Chamber
SCALE FOR Y - DEPTH OF SURGE CHAMBER - FEET.
0 5 10 15 20 25 30 35
5 10 15 20 25
RADIUS OF SURGE CHAMBER - FEET

available will appreciate the great advantage of this new solution offered by Professor Durand in that the size and form of the chamber are definitely and directly determined for a given set of conditions which are chosen to represent the most severe demands liable to be made on the surge reservoir. The selection of a form of acceleration curve on which to base the calculations presents no difficulty after one or two trials.

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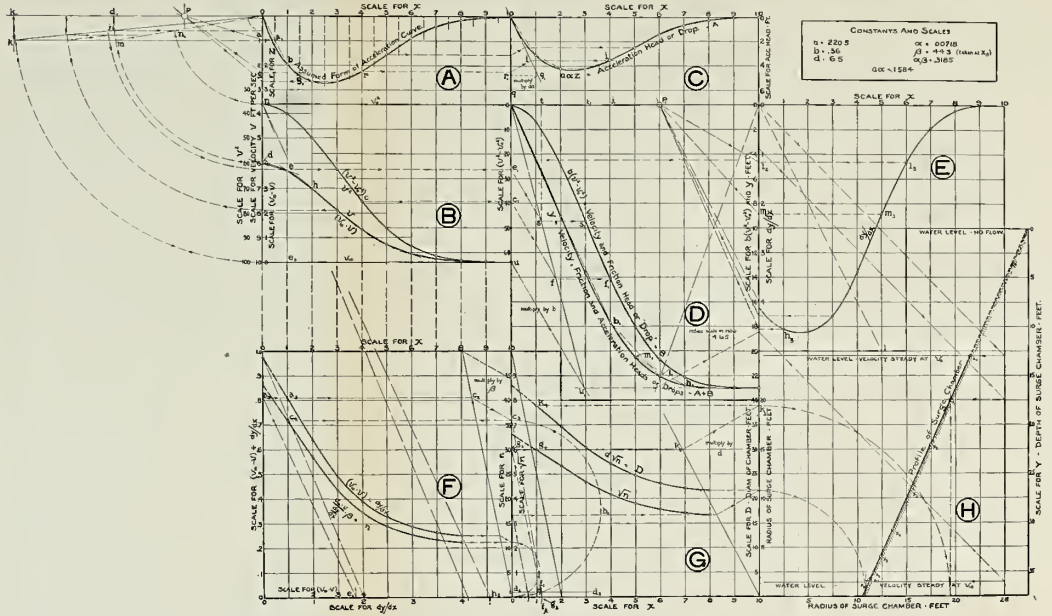
CONSTANTS AND SCALES
a = 220.5
b = .36
d = 6.5
α = .00718
β = 44.3 (Ratio at X₀)
αβ = .3185
aα = .1584
SCALE FOR X
SCALE FOR Y - DEPTH OF SURGE CHAMBER - FEET
RADIUS OF SURGE CHAMBER - FEET
SCALE FOR dy/dx
SCALE FOR (V₀-V) + dy/dx
SCALE FOR (V₀-V)
SCALE FOR V
SCALE FOR V²
SCALE FOR (V₀-V)
SCALE FOR Z
SCALE FOR ACC. HEAD - FT.
SCALE FOR b(V²-V₀²) AND Y - FEET
SCALE FOR D - DIAM OF CHAMBER - FEET
SCALE FOR n
SCALE FOR √n
SCALE FOR √n
SCALE FOR (V₀-V)
SCALE FOR V₀
Assumed Form of Acceleration Curve
Acceleration Head or Drop - A
Velocity and Friction Head or Drop - B
Velocity, Friction and Acceleration Head or Drop - A+B
Profile of Surge Chamber
WATER LEVEL - NO FLOW
WATER LEVEL - VELOCITY STEADY AT V₀
WATER LEVEL
VELOCITY STEADY AT V₀
A
B
C
D
E
F
G
H

FIG. 19

more accuracy the mean ordinate of the curve for the elementary section being treated.

From equation [30] $\alpha\beta = \frac{v_2 - v_1}{R}$, where R = the total area under the curve; $v_2 - v_1$ = the change in velocity = $(v_{10} - v_0)$ in the case under consideration; $\alpha\beta$ = the scale for areas or the product of the scales for acceleration and time. The total area R may be obtained by adding together the ordinates z of the curve, obtaining the mean ordinate and then multiplying the mean by the abscissa of the whole curve; or for the case in question it is numerically equivalent to the sum of the ordinates.

From the known or assumed conditions of demand $(v_{10} - v_0)$ is known. Then we have

$$\alpha\beta = \frac{v_{10} - v_0}{R} = \frac{4}{12.56} = 0.3185$$

and the base of integration PO to be used, giving the velocity curve direct, will be equal to the reciprocal of this or 3.14 for if

x_{10} = units of the variable
z_{mean} = units of the function
$v_{10} - v_0$ = units of the integral

then

$$PO = \frac{x_{10} z_{\text{mean}}}{v_{10} - v_0} = \frac{\text{area under curve}}{\text{units of the integral}} = \frac{12.56}{4} = 3.14$$

The point P having been determined the procedure is as follows: Draw ob, which intersects the mean ordinate between o and 1 at a_1; draw the horizontal a_1a and connect d and P. Starting at d, Diagram B (Fig. 19), the initial point of our velocity curve, draw de parallel to Pd; the intersection of this line with x_1 at e gives the value of v_1 of the velocity curve. Proceeding in the same manner, determine g_1, Diagram A, the point on the mean ordinate between x_1 and x_2, carry the point over to g; draw Pg and through e, Diagram B, draw eh parallel to Pg. This determines the second point or v_2 of the velocity curve. The remaining points of the velocity curve are found in a similar manner.

If now we draw a straight, horizontal line representing the final velocity v_{10}, the distances between this line and the velocity curve will give the values of $(v_{10} - v)$ to be used later in Diagram F, Fig. 19.

It is next desired to obtain the values for v^2. As an example of the method, take the value of v_0, as represented by the ordinate of the point d. With O as a center draw the arc dd_1, at d_1 draw the ver-

tical d_1j, intersecting the diagonal kO at j. This kO is the diagonal of a parallelogram, O_1k_1 whose sides are in the ratio of 10 to 1, so that if we draw through j a line jn_1 parallel to k_1O, we will obtain the value of v_0 to the reduced scale for v^2, so chosen that the values when represented graphically will come within reasonable limits on the paper. From the intersection m of the arc dd_1 and the horizontal k_1, draw the diagonal mO intersecting jn_1 at n_1; then n_1O equals On equals v_0^2 to the reduced scale.

This follows from the relations between similar triangles, where

$$mO : \text{unity} : : n_1O : d_1j$$

or

$$mO \,.\, d_1j = n_1O$$

but

$$d_1 j = \frac{mO}{10}$$

Therefore

$$mO \,.\, \frac{mO}{10} = n_1O$$

or

$$(mO)^2 = 10n_1O$$

That is n_1O represents $(mO)^2$ to 1/10 the scale of velocity. The remaining values of v^2 are found in a like manner, and the curve drawn through the points thus determined.

Drawing the horizontal line representing the value of v_0^2, the distances between this line and the curve give the values of $(v_1^2 - v_0^2)$. If we multiply these quantities by the constant b we will obtain the values of the sum of the velocity and friction heads at the various points, expressed in feet. To do this, use is again made of similar triangles.

Lay off, in Diagram D (Fig. 19) the diagonal ou_1 so that the side $oe_1 : ts : : 1 : b$. Then $ts = b \,.\, oe_1 = b\,(v_1^2 - v_0^2)$. Other points of the curve may be found in a similar manner. Thus by drawing from the respective values of $(v^2 - v_0^2)$ lines parallel to e_1s, the intersection of these lines with the diagonal ou_1 will give the corresponding values of $b\,(v^2 - v_0^2)$.

It is next necessary to obtain the values of the acceleration heads at the same points, and this is done similarly by multiplying the values of z in Diagram A, by aa, a being a constant and a the scale for acceleration using the method of similar triangles just described. Thus the point r, Diagram A, is transferred horizontally to r_1; then in Diagram C, r_1i is drawn parallel to qq_1, which with the diagonal Oq_1

establishes the ratio of 1 to aa. Drawing i_1i we obtain the acceleration head corresponding to the point r. The other values for the acceleration head having been obtained, if we add them to the corresponding ones for the friction and velocity head as obtained in curve B, Diagram D, we will obtain the resulting curve of y, Diagram D. For example $4i_1 + j_1f_1 = j_1b_1$. As the ordinates of the two curves are drawn to the same scale, the summation may be effected by the use of dividers. Having the history of the total drop due to the combined influence of velocity, friction and acceleration it becomes necessary to differentiate this curve.

Choosing a suitable scale for $\frac{dy}{dx}$, the pole point P_1, Diagram E, is determined by taking any values for y and x, as say 20 and 5, draw the diagonal Ol. Then since $\frac{y}{x} = \frac{20}{5} = 4$, through the point 4 of the scale or $\frac{dy}{dx}$ draw a line P_14 parallel to the diagonal Ol; this intersects the base line at P_1 the pole point for the differentiation. To obtain the tangents at the different points of the y curve, take for illustration the point m_1, Diagram D; draw the chord b_1l_1 connecting the intersections of the curve and the ordinates immediately before and after point m_1. Then through the pole point P_1, Diagram E, draw a line P_1m_2 parallel to b_1l_1; Om_2 = the corresponding value of $\frac{dy}{dx}$, draw the horizontal m_2m_3 and the intersection m_3 with the ordinate corresponding to that of m_1, gives the point on the curve for $\frac{dy}{dx}$. The same procedure is followed for the other points.

It is next necessary to obtain the curve giving the results of dividing the values of $(v_{10} - v)$, Diagram B, by the corresponding values of $\frac{dy}{dx}$, Diagram E. This has been done in Diagram F.

On the base line lay off the scales for $(v_{10} - v)$ and for $\frac{dy}{dx}$, the units of which in the present case are as 1 to 2. Adopt a scale for $\frac{v_{10}-v}{\frac{dy}{dx}}$ and locate a point n (falls in Diagram B) such that On is twice unity (for this example) of the scale for $\frac{v_{10}-v}{\frac{dy}{dx}}$, and locate a point n (falls

in Diagram B) such that On is twice unity (for this example) of the scale for $\frac{v_{10}-v}{\frac{dy}{dx}}$. The dimsion is accomplished as follows:

With the dividers take off the distance ee_3, Diagram B, being the value of $(v_{10} - v)$ for ordinate 1; lay this off on the scale for $(v_{10} - v)$, Diagram F, so that $oe_2 = ee_3$ of Diagram B. On the scale for $\frac{dy}{dx}$ lay off $Oh_2 = 1h_3$ of Diagram E, draw nh_2 and through e_2 parallel to nh_2 draw e_2a_2, and then Oa_2 will be the value of $\frac{v_{10} - v}{\frac{dy}{dx}}$ for ordinate 1. Drawing a horizontal a_2a_3 till it intersects the ordinate 1 at a_3, we have a point on the curve and so on for the other points.

The next step is to multiply these values by β, the scale for time, the result being the value of n. The method followed is the same as previously described and is indicated on the diagram.

To obtain the values for the $\sqrt{n}$, take point c_4 Diagram F, as an example, draw c_4c_3 and upon Oc_3 as a diameter construct the semicircle c_2b_2O, Diagram G, draw d_3d_2 parallel to the base through unity of the scale for n, this intersects the semicircle at b_3. Then $ob_3 = \sqrt{oc_3}$ for from the similar triangles c_3b_3O and Ob_3d_3

$$Oc_3 : Ob_3 :: Ob_3 : 1$$

or

$$(Ob_3)^2 = Oc_3$$

Therefore

$$Ob_3 = \sqrt{Oc_3}$$

As the resulting values of $\sqrt{n}$ are rather small referred to the scale of n a scale for $\sqrt{n}$ has been chosen so that the unit value is five times that for n. Having the values of $\sqrt{n}$ to the scale of n we may transfer them in the ratio of 5 to 1 as follows. Draw the diagonal from the point $x = 2$ to $\sqrt{n} = 10$. Then with O as a center and ob_3 as a radius strike the arc f_2f_3, cutting the base at g_2. From g_2 draw g_2g_3 parallel to the diagonal 10–2. The intersection of the horizontal g_3g_4 with the ordinate 1 gives a point on the $\sqrt{n}$ curve. Finding the remaining points and multiplying the values by d (the equivalent diameter of the conduit) as indicated, we obtain the corresponding values D for the diameter of the surge chamber, or on a scale of 2 to 1, these values will represent the radii.

Transferring by rotation these values of the radii to form the abscissae of Diagram H (Fig. 19) and from Diagram D transferring with change of scale (to correspond to that of the radii) the values of y to which are also added the initial level of the water in the reservoir at velocity v, we have the ordinates for Diagram H. The corresponding values of Y and r when plotted give the profile of the surge chamber required to meet the assumed conditions for demand and acceleration curve adopted in Diagram A.

With reference to the selection of a form of acceleration curve, as Professor Durand has already stated the typical form given by him will fit approximately widely dissimilar cases. This may be illustrated by the application of the Los Angeles type of curve to the Snow Mountain and Colgate projects.

The data relative to the two cases are as follows:

	Snow Mt., ft.	Colgate, ft.
Total length of conduit.....	7097	26,200
Equivalent diameter of conduit.....	6.5	8

Referring to Fig. 20, curve M is one of the typical Los Angeles curves with the values of x and z taken as used in that case. The resulting profile of the surge chambers developed from this curve for both the Colgate and Snow Mountain projects is plotted below. It will be seen that though for widely differing cases the general outline of the chambers is the same. A second form of acceleration, curve N, shown by the dotted line, results in a straight line profile for both reservoirs, as indicated by the corresponding profiles.

In order that the differences between curves M and N may be more easily compared and their effect upon the profiles more correctly judged, the ordinates of both having been arbitrarily chosen, curve M has been reproduced at M', with the ordinates so proportioned that the area under curve M' is equal to the area under curve N. The scales for area, $\alpha\beta$, are then the same; and were the scales for time, β_1, also the same in both cases the two curves would be exactly comparable. For the same project the time scales are very close, so that deductions may be drawn from the curves plotted in this manner without sensible error.

The curve M' may be determined graphically from curve M by plotting it to a new base of integration equal to that for N, or it may be determined numerically from the relation that

$$z^1 : z :: \text{area under } N : \text{area under } M$$

or

$$z' = z \frac{\text{area under } N}{\text{area under } M}$$

where

z is an ordinate of the curve M

z' is the corresponding ordinate of curve M'

The chairman of the meeting asked Professor Durand if he had considered the effect of a chamber not open to the atmosphere at the

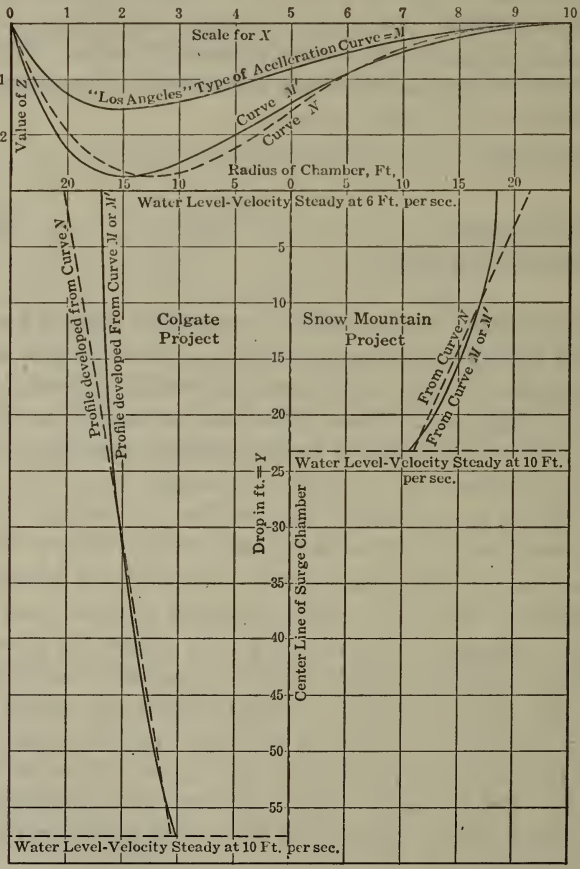

FIG. 20 ACCELERATION CURVES AND RESULTING PROFILES FOR SURGE CHAMBERS

upper end, having atmospheric pressure at the point of uniform velocity for a given set of conditions, and then varying those conditions so as to produce a pressure in the surge chamber above the water

greater than atmospheric pressure; or if it fell below the level for a constant velocity for an assumed set of conditions, there should be a vacuum, in other words, a pressure tending to accelerate or retard flow, what would be the general effect of such a set of conditions?

LARS JORGENSEN.[1] The value of having a surge tank on a long closed conduit is generally realized. The problem is, however, somewhat complex and any attempt to throw new light upon it should be welcome to the profession. Professor Durand's analysis is interesting and instructive. In starting out from an assumed acceleration curve he arrives at a certain shape and size of the surge basin. For mechanical reasons a conical tank is evidently the best if it has to be cut out of solid rock and lined with concrete, and therefore it may be necessary in many cases to make it conical. For other reasons it would, however, be better to have the basin wide at the bottom and to have it larger than absolutely necessary to keep down the amount of surge to a value considered small enough. A conical tank, or any other shaped tank barely large enough to keep down the amount of surge will answer the purpose it is intended for, but by excavating somewhat more and in the right place it is surprising to see how much the average effective head on the turbines can be increased, in places where the gravity flow line is long, closed of course, and on a steep grade. In the example mentioned by Professor Durand, where the minimum and maximum flow through the power conduit are about 600 sec-ft. and 1000 sec-ft. respectively, it would probably be out of the question to make the surge tank larger than absolutely necessary to take care of the surge, if no natural depression could be found in the vicinity of the beginning of the pressure pipe proper adapted for reservoir use. This was probably not at hand, otherwise Professor Durand would have used it. The closed conduit is long, but the grade is so small that the average effective head possible to gain by having a small regulator to help take care of the peak and to keep the velocity flow more constant throughout the 24 hours would not be enough to compensate for digging such a basin which would have to accommodate about 6,000,000 cu. ft. of water or more (400 sec. per ft. during 4 hours or more).

On developments having a less amount of water, the conditions change in favor of having a surge tank larger than just necessary to keep down surges. The smaller the amount of water to be handled, the steeper the grade of the flow conduit has to be for commercial reasons. In that case, however, we have a good chance of recover-

[1] Electric and Hydraulic Engineer, 1406 Chronicle Bldg., San Francisco, Cal.

ing some of this fall by making the surge tank large enough to cut off the peak of the velocity (during the peak load), resulting in a more than proportional saving in head loss. As the loss in the conduit in this case would be more nearly constant over the 24 hours, we would be able to keep a fairly constant head on the turbines, except toward the very end of the peak load, and they are therefore capable of keeping up their efficiency at a time when that efficiency is most needed.

Suppose we had 200 sec-ft. average flow in the 41,000 ft. of tunnel, our dimensions would be about as follows: 6 ft. inside diameter concrete lined tunnel on a gradient of 0.00424, which corresponds to $n = 0.012$ in Cutter's formula. The tunnel is made circular in cross-section, as this is by far the best and safest form for any lined tunnel of large size for water carrying purposes, both from a mechanical and hydraulic standpoint, and it is not much, if any harder to excavate. If the minimum and maximum demand was 150 and 250 sec-ft. respectively (that is dividing the existing conditions in the case cited by Professor Durand by four, thereby reaching conditions which correspond more nearly to an average size development in California, Nevada, etc.), then the small receiving reservoir or surge basin should be made large enough to store enough water at times when the demand of the pressure pipe proper is below 200 sec-ft. to supply the demand when this is above 200 sec-ft. which would require a storage of about 1,500,000 cu. ft. (more or less depending upon duration of peak load).

For the sake of economics we cannot afford to carry the top level of our small regulator up to the maximum static level; if we did, we would need a storage of several times 1,500,000 cu. ft. The top level of the surge basin should be the amount of the friction head corresponding to 150 sec-ft. flowing below the static elevation, or in this case (one foot slope per 1000 ft.) 41 ft. below. Should the demand happen to be less than 150 sec-ft. after the reservoir was filled up, the water would simply spill over, which in most cases would not be objectionable, or else regulated from the upper end. Friction loss in a 6 ft. diameter conduit, 41,000 ft. long, when carrying 200 sec-ft. about 74 ft., and when carrying 250 sec-ft. about 116 ft., the difference, 42 ft. (5 per cent of head), average would be gained over the peak, more at the beginning and less at the end. Right here it is seen that in order to keep up the head towards the end of the peak period it is necessary to have the bottom of the reservoir wide. At the beginning of the peak the small reservoir supplies much more than its share until the level has dropped and the water in the conduit has speeded

up, therefore it would be better for the purpose to have the reservoir narrow at the top and wide at the bottom. As this would hardly be practical, the only remedy is to make the reservoir somewhat larger than 1,500,000 cu. ft. This would only be necessary where water is drawn from a large reservoir, where the top level of the water is not constant. An error made in the calculation of the conduit could afterwards be more or less corrected by making the regulator of larger capacity. Besides increasing the average head, at least over the peak, about 42 ft. and raising the efficiency of the turbines, this small reservoir cuts down the static head for which the pressure pipe must be designed 41 ft. (about 5 per cent of head).

Taking everything into consideration, it is easily seen that whenever conditions are right it may pay to spend a great amount of money in constructing a small regulator in place of a surge tank. The small regulator will perform all of the duties of the surge tank to perfection, and having a larger area, will keep the surge smaller as the time for acceleration and retardation has been extended to hours instead of minutes.

The Author. In selecting a form of acceleration curve for any given case the first point to be considered is the program as regards acceleration and velocity which it is desired the water should follow. Thus if the movement is to be substantially non-oscillatory, the form of the curve at the end must be similar to that of Fig. 5, while if surging may be permitted the form may approach that of Fig. 2. In general it is found that all typical cases present acceleration curves of closely similar characteristics. Thus there is always a rapid rise at the beginning of the time interval during which the acceleration reaches a maximum, followed by a decrease with or without a point of inflection and a gradual approach to the time axis, according as the program is to be non-oscillatory or otherwise. It must also be remembered that, in accordance with the present method of attack, the program for the water is to be fixed in advance and we are then to accept within limits the resultant dimensions and form of chamber. The two causes for revision will be therefore either a resulting chamber unsuited in size to local or economic conditions, or else a form of profile undesirable or impracticable in character.

It will quickly appear in the study of these problems that there are two things upon which will depend the relation between the form of the curve and the profile of the resultant chamber, viz.,

the value of β the time interval, and the actual form of the acceleration curve. A study of these relations along the lines indicated in Pars. 83-88 will usually serve quickly to indicate such changes as may be needed in order to modify the resultant form or dimensions of chamber along any desired line.

Regarding the use of a closed chamber with air control I may say that while such method is undoubtedly practicable in many cases, I had not considered it with reference to the particular case under discussion, primarily on account of the very large dimensions of the chamber which would be required. As pointed out by Johnson in the paper to which reference has previously been made,[1] the use of an air chamber will result in a marked reduction in the vertical dimensions of the chamber, but the transverse dimensions would still remain, for the present case, large beyond practicable limits. Also, as pointed out by the same author the conditions as regards regulation will not be as satisfactory, unless at least the differential feature as proposed by him be included in the design.

Replying to Mr. Jorgensen, it is perhaps sufficient to say that he is entirely correct in assuming that the topographical conditions in the particular case under discussion are not such as would permit the installation of a regulating reservoir in the usual sense of the term. Such a reservoir, if of size to carry across the peak load will, of course, exercise a most helpful control over the loss of head at high velocities. In effect it will serve to eliminate the higher range of velocities in the long conduit and restrict them simply to the short conduit between reservoir and power house. These possibilities are well shown by the case which Mr. Jorgensen assumes for illustration.

It may be well to add here as a further variant to the method outlined in the paper that since its preparation it has been found possible to proceed quite satisfactorily, so far as the computations are concerned, by going one step further back in the assumed curve and specifying the differential of the acceleration curve instead of the acceleration curve itself.

Reference to the equations of the paper and to the outline of method will readily show that by the use of equations [22], [23], [24] and [25], the problem may be solved by making two integrations in succession, the first for the acceleration curve and the second for the veloc-

[1] Trans. Am. Soc. M. E. vol 30, p. 4430

ity curve, instead of one integration and one differentiation as used in the examples of the paper. It must be admitted, however, that a suitable form of curve is not as readily assumed in the case of this

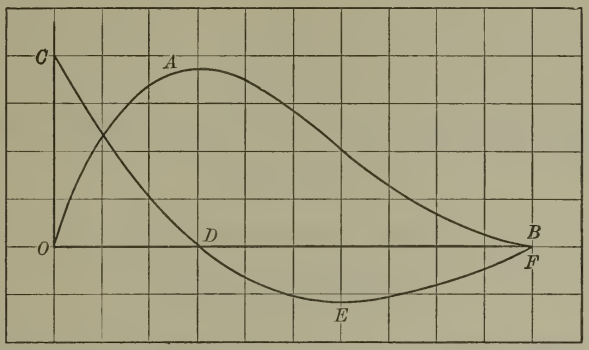

FIG. 21 TYPICAL FORM OF ACCELERATION CURVE WITH ITS DIFFERENTIAL

method as in that of the paper. In Fig. 21 is shown a typical form of initial curve together with its integral, the acceleration curve.

This possibility of variation in mode of procedure is here suggested without further development of detail.

No. 1354

SPEED REGULATION IN HYDRO-ELECTRIC PLANTS

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Member of the Society

Although speed regulation of the turbine unit is one of the most important features in a successfully operating hydro-electric plant under modern conditions, it rarely receives more consideration in the design of a plant than to provide for the installation of a governor of some standard type.

2 The result is that a hydraulic turbine governor may be praised in one plant and condemned in another almost exactly similar, and the engineer contemplating such an installation often finds after the most thorough investigation that he can obtain recommendation and condemnation of every governor on the market.

3 The reason for this is invariably evident from an analysis of the various features of the plants regarding which information may be at hand, and of the original purpose for which any particular governor was designed.

4 In many plants practically no regulation is required, either because the prime movers operate under ideal conditions, or, although the conditions are not so good or perhaps even quite bad, the loads are almost constant. The most severe tests to which the governor is put in such plants is the shutting down of a turbine in the case of a short circuit, and it is not surprising that the reports of governor operation from such plants are good. On the other hand, it often happens that governors are installed where the conditions for speed regulation are severe, or where the instantaneous or constant load changes are a large percentage of the unit or plant capacity, and if the subject was not given proper consideration in the

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design of the plant, it is not strange that bad reports are given of governors which in these other plants are highly satisfactory.

5 In the present state of governor development, there is no reason why commercial regulation should not be obtained in every hydro-electric plant even if not always equal to that obtained in modern steam plants, it should at least be very nearly as satisfactory. This, however, can not always be accomplished by the installation of a well-designed governor alone, as the governor can perform only certain of the many necessary functions, no matter how perfect it is. In other words, the several parts of the whole hydro-electric plant must be put in such relationship that the governor, if properly designed, can perform the functions necessary for good commercial regulation.

6 If the plant has been arranged as above stated, very little difficulty will be experienced with any one of the governors which have been put on the market by a number of reputable manufacturers in this country. There are cases, however, where the governor of one particular type or design may be more adaptable than the rest or some of the rest.

7 The sale of a governor is seldom left to its designer, who may know its limits and possibilities, and it is therefore important that the guarantee made by the manufacturer or his agent be carefully checked by the engineer designing the plant in which the governor is to be installed, even though the governor be the best on the market.

8 Certain well-designed governors are in bad repute with some engineers, simply because the manufacturer or his agents were so eager to make a sale, that the question of adaptability, or the conditions under which the governor was to operate were overlooked, or if considered at all, a chance was taken that the plant would be an easy one to operate.

9 It is the purpose of this paper to give formulae by means of which the probable speed regulation can be computed for any set of conditions, and to make suggestions which may be helpful in arriving at a satisfactory proportioning of parts for good commercial results.

10 The regulation of hydro-electric units as now accomplished is one of degree only, since a departure from normal speed is necessary before the governor can act. Since the immediate effect of the gate motion is the opposite of that intended, the speed will depart still further from the normal. This tends to cause the governor to move the gate too far, with the result that the speed will not only return to normal as soon as the inertia of the water and the rotating parts is overcome, but may rush far beyond normal in the op-

posite direction. The obvious tendency is thus to cause the speed to oscillate above and below normal (hunting or racing) to the almost complete destruction of speed regulation.

11 A successful governor must therefore anticipate the effect of any gate movement. It must move the gate to, or only slightly beyond, the position which will give normal speed when readjustment to uniform flow in the flume or penstock has taken place. A governor with this property or quality is commonly said to be a "dead-beat" governor. The expedient used for this purpose on hydraulic governors is called a compensating device, which may be either simple or compound.

REGULATION OF OPEN-FLUME TURBINES

12 For the regulation of turbines in open flumes, where the velocity of approach does not exceed 2 to 3 ft. per sec., a gate movement only slightly increases or decreases the head, so that it may be considered that the immediate effect is the desired one, namely, to increase the flow of water for gate opening and decrease it for gate closing, so that the inertia of the water may be disregarded, consideration for the inertia of the rotating masses only being necessary. Thus in plants where turbines are used in open flume the danger of hunting or racing is practically eliminated, and by choosing the proper value for the rotating masses almost any desired speed regulation can be obtained. However, this is true only where the length of the draft tube is short compared with the head. The water in the draft tube must be accelerated or retarded at each change of gate opening and its kinetic energy changed at the expense of the power output in exactly the same manner as that in the penstock or flume. The effect is a tendency towards hunting or racing and it may be found that regulation with a governor is absolutely impossible at part load. In any case where a draft tube is used it must be included in all calculations. If the draft tube is long, and the velocity in it comparatively high, a quick closure of the turbine gates may cause the water in the draft tube to run away from the wheel, actually creating a vacuum, and then return again causing a destructive blow against the turbine. In such case, regulation of any kind is entirely out of the question.

13 From what has just been said, it is evident that with turbines in open flume, if there is a suitable velocity of approach in the flume and the draft tube is of proper dimensions, the regulation is a question of the inertia of the rotating masses or flywheel effect.

FLYWHEEL EFFECT

14 Flywheel effect is the capacity of a rotating mass to store or supply energy. This effect is expressed in pounds at 1-ft. radius, and is equal to the moment of inertia of a mass, that is, the weight by the square of the radius of gyration, or pounds feet squared. Modern alternating-current generator rotors are built in the shape of a common flywheel and usually have sufficient flywheel effect to obtain satisfactory regulation. The flywheel effect of the rotating parts of a turbine varies with different types and usually is not considered; but it should be made a matter of care for large low-head multiplex and large diameter impulse turbines. In smaller turbines it may be regarded simply as introducing a factor of safety.

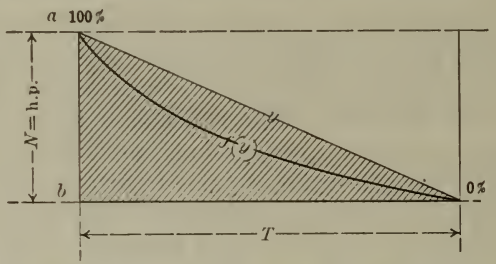


FIG. 1 ENERGY OF ROTATING MASSES GIVEN OUT IN TIME T

15 In turbines of very large diameter and in high-power reaction turbines, the weight of the water in the runner must also be considered to make the calculations of any value, as its effect is exactly the same as that of any other rotating mass, and may be a large percentage of the total rotating mass. With geared turbines, the effect of all gears and shafting should be considered.

16 Where turbine-driven direct-current exciters are used in a large hydro-electric plant, their possible regulation should be carefully considered, as the flywheel effect of these exciter generators is usually small and an additional separate flywheel may prove necessary.

17 When purchasing generators, a comparison of the flywheel effect should be made, as they are usually built as light and compact as possible. Additional flywheel effect within certain limits can generally be obtained at a very small cost in alternating-current generator rotors, and it will ordinarily prove cheaper and more convenient to do this than to add a separate flywheel.

18 If a separate flywheel must be added, its weight for a given moment of inertia or flywheel effect depends upon the permissible

peripheral speed. In calculating a flywheel, the runaway speed (for full gate opening and friction load) must be considered and is really the determining factor. For an impulse turbine this runaway speed will be about 90 per cent above normal. For the standard types of reaction runners, *A*, *B*, *C*, *D*, *E*, and *F* (See Appendix, Table 1), the runaway speed will be respectively 70%, 65%, 60%, 55%, 50% and 45% above normal. These are the maximum speeds and they may be appreciably less, depending upon the setting of the turbine and the type of bearings used. For low speeds, flywheels of the engine type may be used, but for high speeds they should be of the disc type, turned all over and carefully balanced. The maximum peripheral speed for solid cast iron disc wheels should not exceed 100 ft. per sec. and for cast-steel wheels of the same type, it should not exceed 200 ft. per sec.

19 Cast-steel wheels are usually more convenient on account of their smaller size for the same flywheel effect, and cheaper because less additional bearing surface is required, and the greater cost per pound of material, if any, is offset by the lighter weight, the flywheel effect varying with the square of the radius of gyration.

20 The regulation due to any given flywheel effect can be computed by means of the formula

$$d = \frac{800{,}000 \times N \times T}{n^2 \times Wr^2}$$

It must usually be modified, however, and to show clearly how this comes about, the derivation of this formula is given herewith, followed by the symbols for the quantities used.

21 The energy of a rotating mass is $\frac{Mv^2}{2}$. If the peripheral speed of the mass M is reduced from v_2 to v_1 it will lose energy amounting to

$$\frac{M\ (v_2^2 - v_1^2)}{2}$$

Factoring $(v_2^2 - v_1^2)$

$$(v_2^2 - v_1^2) = (v_2 + v_1)\ (v_2 - v_1)$$

Substituting in the above and multiplying by $\frac{v}{v}$ we have

$$\text{Energy lost} = \frac{Mv\ (v_2 + v_1)}{2} \times \frac{(v_2 - v_1)}{v}$$

but

$$\frac{v_2 + v_1}{2} = \text{average velocity} = v$$

and

$$\frac{v_2 - v_1}{v} = \text{relative change in velocity} \equiv d$$

(d is also called the percentage of ununiformity).

22 Substituting the values of v and d in the above equation of energy lost, we have

$$\text{Energy lost} = Mvvd = Mv^2d$$

or

$$M\frac{(v_2^2 - v_1^2)}{2} = Mv^2d$$

23 Referring to diagram (Fig. 1), let ordinates represent horse-power and abscissae represent time. Then the energy, $\frac{NT}{2}$ is that which must represent the amount given out by the rotating masses, after the turbine gates have been closed off, and T is the time required to give it up. That is, if the total load is taken off suddenly, the power would drop from a to b, but on account of the rotating masses this would not happen instantaneously, but would take a certain time T and the effect of the masses is equal to the area below the line y and if this is a straight line the area $= \frac{NT}{2}$. Then

$$\frac{NT}{2} \times 550 = \frac{W}{g}\left(\frac{\pi^2r^2n^2d \times 4}{60 \times 60}\right)$$

where

$$M = \frac{W}{g}$$

and

$$v = \frac{2\,\pi rn}{60}$$

Substituting numerical values and transposing, we have

$$Wr^2 = \frac{NT}{n^2d}\left(\frac{60 \times 60 \times 32.2 \times 550}{2 \times 3.14 \times 3.14 \times 4}\right) = \frac{NT}{n^2d}800{,}000 \ldots\ldots [1]$$

and

$$T = \frac{Wr^2n^2d}{N \times 800{,}000} \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

also

$$d = \frac{800{,}000 \times N \times T}{n^2 \times Wr^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

which is the formula as first given, and

Wr^2 = moment of inertia of the rotating masses in lb. ft. squared

T = time in seconds to close or open gates by the governor

n = revolutions per minute

$$d = \frac{v_2 - v_1}{v} = \frac{n_2 - n_1}{n} = \text{the relative change of speed}$$

$$= \frac{(D\,w)}{w}$$

$$w = \frac{\pi n}{30} = \text{angular velocity}$$

N = horsepower

Correction. In deriving the above formula, it was assumed that the line y was a straight line and the area under it was equal to $\frac{NT}{2}$, under the assumption that the total load was thrown off.

24 However, the friction load of the turbine and generator remains, and the actual area represented will be that under a curve $f(y)$. The effect of this friction load will, of course, differ with the various types of turbines used, but the original formula may be modified safely as follows, the curve $f(y)$ being approximately a parabola:

$$d' = 0.8\,d \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [4]$$

25 Another cause which should be taken into consideration, providing the turbine is properly designed so as to give its maximum efficiency at normal or synchronous speed, is that the efficiency is reduced with either increasing or decreasing speed, which has the tendency to reduce still further the area under the curve $f(y)$. The amount reduced varies with the type of runner, being greatest for a high-speed high-power runner, and least for a low-speed low-power runner, since, if correctly designed and therefore following the laws of hydraulics, the change in efficiency will be greater for a high-speed than

for a low-speed runner for the same per cent of variation in speed. The formula modifying d' for this cause, is as follows:

$$d_{\text{eff}} = \frac{d'}{1 + \frac{d'}{\frac{n_1}{n_0} - 1}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

where $\frac{n_1}{n_0}$ varies from 1.8 for a runner of low speed to 1.3 for a runner of high speed.

26 Referring to Table 1, giving standard runners types A to F (see Appendix), designed under the writer's direction for the Allis-Chalmers Company, the values of $\frac{n_1}{n_0}$ are as follows:

Type of Runner.....	A	B	C	D	E	F
Value of $\frac{n_1}{n_0}$	1.8	1.7	1.6	1.5	1.4	1.3
Specific speed $k = \frac{\text{r.p.m.}}{H}\sqrt{\frac{\text{h.p.}}{\sqrt{H}}}$	13.55	20.3	29.4	40.7	(49.7 −62.8)	(70.97 −83.78)

27 The specific speed is that speed in revolutions per minute of a turbine runner, diminished in all dimensions to such an extent as to develop 1 h.p. when working under the head $H = 1$ ft. This value is called type characteristic by Professor Zowski in his article in the Michigan Technic of June 1908, The American High Speed Runners for Water Turbines.

28 To simplify calculations, this table will prove efficient:

Type of Runner....	A	B	C	D	E	F
$0.8x$ in $d_{\text{eff}} = 0.8xd$	0.714	0.703	0.69	0.671	0.645	0.606

29 For impulse turbines the same d_{eff} as for type A reaction runner should be used.

30 To find the speed variation for any other than a full load change, the following must be considered: If T is the regulating time for the total governor stroke or 100 per cent load of the turbine, it would appear that for 50 per cent load or stroke, the regulating time would be $\frac{T}{2}$, and for 25 per cent, $\frac{T}{4}$, etc.

REGULATION WITH MECHANICAL GOVERNORS

31 As far as the stroke is concerned, this is nearly true with a mechanical governor, and the speed variations for part load changes can be calculated as follows:

32 The formula for d which we have given, may be transformed to appear:

$$d = \frac{Z^2}{2\left[\frac{2\left(\frac{mv^2}{2}\right)}{550\ NT}\right]} \left[1 - \frac{2\ aZ}{3\left[\frac{2\left(\frac{mv^2}{2}\right)}{550\ NT}\right]}\right]$$

in which the new symbols Z and a are, respectively, the change in load, and the remaining load.

33 In the foregoing formula, if we let

$$S = \frac{1}{2\left[\frac{2\left(\frac{mv^2}{2}\right)}{550\ NT}\right]}$$

it will appear as follows:

$$d = SZ^2\left(1 - S\ \frac{4\ aZ}{3}\right) \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

34 From this formula, we can now compute d for various part loads, since S, as may be readily seen, is d as figured from the original formula for total change of load, or $S = d'_{\text{eff}}$ in the above formula, and for load changes of 25%, 50% and 75% we have the following:

a For 25%, $Z = 0.25$ and $a = 0.75$
$d'\ 25\% = d'_{\text{eff}} \times 0.625\ (1 - d_{\text{eff}} \times 0.25)$

b For 50%, $Z = 0.50$ and $a = 0.50$
$d'\ 50\% = d'_{\text{eff}} \times 0.25\ \ (1 - d'_{\text{eff}} \times 0.333)$

c For 75%, $Z = 0.75$ and $a = 0.25$
$d'\ 75\% = d'_{\text{eff}} \times 0.56\ (1 - d'_{\text{eff}} \times 0.25)$

35 The results of these computations plotted appear as in Fig. 2, in the line marked a.

36 Line a correctly represents the speed variation under the assumption that the governor acts instantaneously upon a speed change

however, a certain speed change must occur before the governor can act on account of the insensitiveness of the flyballs, and the actual speed changes are more correctly represented by a line *b*, midway between the curve *a*, and a straight line *c*, as shown..

REGULATION WITH HYDRAULIC GOVERNORS

37 For a hydraulic governor, an entirely different line of reasoning must be followed. My own observations are supported by those of others who claim that the regulating time for all gate openings is nearly constant and equal to that of the total gate opening for a hydraulic governor.

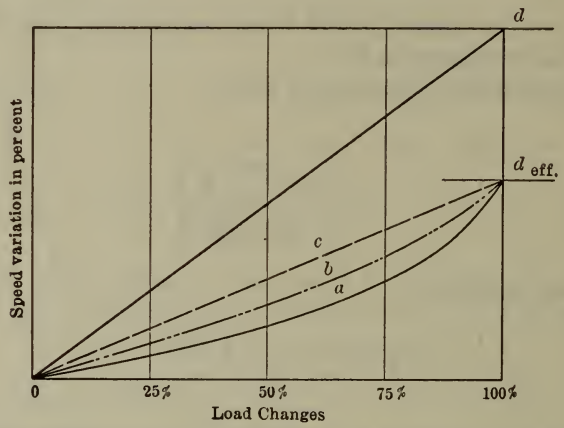


FIG. 2 APPARENT AND ACTUAL REGULATION OF MECHANICAL GOVERNOR

38 This is due largely to the throttling effect of the regulating valve and the time required to overcome the inertia of the regulating masses. That this fact is not detrimental to the speed regulation can be seen from what follows, and that it is of large benefit to the problem of pressure variations and therefore indirectly again to regulation, will be seen from a study of the following discussion on pressure variations.

39 From experience and tests, it has been found that for hydraulic governors the curve *a*, shown under partial load speed variation for mechanical governors, must be modified as shown in Fig. 3.

40 The curve *d* is an arc of a circle drawn tangent to the line *od* which is the line of apparent regulation, and through the point d_{eff} which is the actual value of d' as modified for various causes as given.

41 It would appear from a comparison of the regulation curves

given that the speed variation would be less for partial loads with the mechanical governor than with the hydraulic governor. This, however, is not true for practical purposes, since the regulating time for any but very small mechanical governors is, of necessity, very great compared with the hydraulic governor, the regulating time of which can practically be made as short as desired for any size. From an investigation of the original formula for d, it can be seen that d varies directly with T, and therefore, if we can use a shorter time T, with the hydraulic governor than with the mechanical, it is seen that the apparent advantage can be overcome many times, due to the very short time T, which is possible.

42 A mechanical governor can perform only a definite amount of work per unit of time, depending upon the size and strength of the

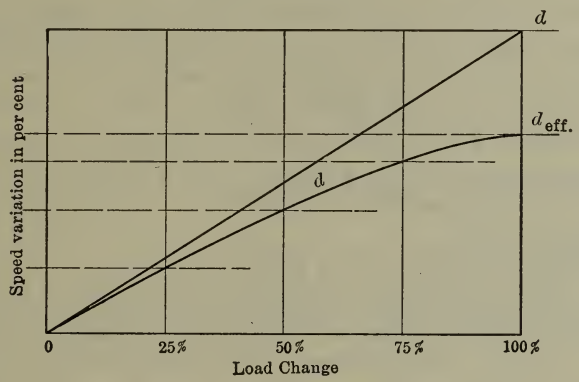


FIG. 3 APPARENT AND ACTUAL REGULATION OF HYDRAULIC GOVERNORS

driving belt or gears which must transmit the power of the governor. If for instance, a mechanical governor is capable, due to the strength of its parts, of transmitting 1 h.p. = 550 ft. lb. per sec., and the work required to regulate a turbine is 11,000 ft. lb., then the regulating time must be $\frac{11{,}000}{550} = 20$ sec.

43 The work which a hydraulic governor is capable of doing is a function of the size of the regulating cylinder, and the time in which it can do this work depends upon the time in which sufficient oil can be brought to the cylinder. This time again depends upon the pressure in the pressure tank and the size and discharge coefficient of the oil pipes and the regulating valve.

REGULATION OF ENCASED TURBINES

44 If the water is conducted to the turbine in a long penstock (encased turbine) a large amount of energy is stored in the moving column of water and a change in its velocity involves a change in its kinetic energy and produces pressure variations, which may, if an attempt is made at too rapid regulation, leave the turbine deficient in energy when increased power is desired, or when the power is decreased, may produce such shocks as will seriously affect regulation, or perhaps result in serious injury to the penstock or turbine.

PRESSURE VARIATIONS

45 With open-flume turbines, these pressure variations may be disregarded, but with encased turbines they must always be carefully considered. These variations can never be entirely done away with in practical cases, but they can be minimized to almost any extent by proportioning properly the various parts of a plant. The solution of such a problem is generally determined by commercial considerations, and a balance must be struck between cost of penstocks and cost of flywheels, if it is impossible or undesirable to modify conditions with other appurtenances.

46 Pressure variations, to a greater or less extent, occur in every practical turbine installation, whenever the amount of water which flows to the turbine is varied, as in the case of a change of load. However, for practical considerations, as far as regulation of speed with governors is concerned, these variations can be considerable before they will have any appreciable effect on the speed regulation.

47 In modern plants, where the tendency usually is to develop as much power and head as possible in a single plant, it frequently becomes necessary to carry the water for some distance in flumes, pipe lines or tunnels in order to concentrate the head on the plant. The length of the conduits often becomes many times greater than the head, and as such long conduits become a large factor in the cost of the plant, the natural tendency is to reduce the size of conduit and increase the velocity of the water flowing in or through it. However, head, length of conduit, and velocity of water in the conduit, are the three principal factors that produce pressure variations, and their relation to each other, as well as their effect on regulation of speed, becomes a very important problem in such plants.

48 In such a plant, the effect of the flywheel masses is the same as before, and the same theory and formulae apply to it. However,

instead of a constant head acting on the turbine, we will have a variable head, which may become greater or less than the normal, depending upon the change of load. This variation in head tends to increase or decrease the amount of power which would be developed for the same gate opening at normal or constant head, and the tendency of the flywheel effect to keep the speed constant is lessened by the amount of increased or deficient power so caused.

49 As calculated from the foregoing formula, "d effective" must be modified by the following formulae due to the causes above mentioned.

For closing gates

$$d''' = d_{\text{eff}} \left(1 + \frac{(DH)}{H}\right)^{\frac{3}{2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [8]$$

For opening gates

$$d''' = \frac{d_{\text{eff}}}{\left(1 - \frac{(DH)}{H}\right)^{\frac{3}{2}}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [9]$$

The derivation of these formulae and the theory leading up to them are herewith given.

50 We know, from the fundamental axioms of mathematical physics, that a given mass acted upon by a variable force in an interval of time is accelerated or decelerated. This relation is given by the following fundamental equation

$$M\frac{\mathrm{d}v}{\mathrm{d}t} = P$$

and

$$M\mathrm{d}v = P\mathrm{d}t$$

51 In the problem relating to speed regulation we always deal with a fixed time T, and a variation in the velocity of the water brought about in the time T, which is identical with a pressure variation. We may, therefore, write

$$M(Dv) = (DP)T$$

where

$$M = \text{mass} = \frac{W}{g}$$

in which

W = weight in lb.

$g = 32.2$ = acceleration of gravity

and

P = pressure = Hy

in which

$\frac{P}{y} = (DH)$, where y = specific weight

Let L represent the length of a pipe line or penstock and F its area. Then

$$M = \frac{LFy}{g}$$

$$P = FP = Fy\,(DH)$$

And substituting in the equation $M(Dv) = (DP)T$, we have

$$\frac{LFy}{g}(Dv) = Fy\ (DH)T$$

Transposing

$$(DH) = \frac{L(Dv)}{gT} \quad \ldots\ldots\ldots\ldots\ldots\ldots [10]$$

or the actual increase or decrease of head for certain regulating time T, and a variation of the velocity of the water of (Dv).

52 This formula, to express the result in per cent of the total head, will read

$$\frac{(DH)}{H} = \frac{L(Dv)}{gTH} \quad \ldots\ldots\ldots\ldots\ldots\ldots [11]$$

and for both increase and decrease it will read

$$\frac{(DH)}{H} = \pm \frac{L(Dv)}{HgT} \quad \ldots\ldots\ldots\ldots\ldots\ldots [12]$$

where $+$ is for closing and $-$ is for opening gates.

53 From this formula we learn that the pressure variations vary directly with the length of the penstock and the velocity of the water in it, and inversely as the head and the time during which the original velocity is varied by (Dv).

54 If we assume an allowable pressure variation, $\frac{(DH)}{H}$, the regulating time can be found from a transposition of the above formula, as follows:

$$T = \pm \frac{L(Dv)}{Hg\frac{(D\,H)}{H}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [13]$$

As an example, let

$L = 500$ ft.
$H = 100$ ft.
$v_0 = 100$ ft. per sec. for $T = 0$
$v_1 = 5$ ft. per sec. for $T = 2$ sec.
$(Dv) = v_0 - v_1 = 5$ ft. per sec.

Then

$$\frac{(DH)}{H} = \pm \frac{500 \times 5}{100 \times 32.2 \times 5} = \pm 0.39 = \pm 15.5\%$$

which shows that for a change in penstock velocity of 5 ft. per sec. the pressure would increase or decrease 15.5%.

55 Where a pressure variation not to exceed 10% is permissible, $\frac{(DH)}{H} = 0.10$, and the corresponding regulating time for the same conditions must be

$$T = \pm \frac{500 \times 5}{100 \times 32.2 \times 0.10} = 7.8 \text{ sec.}$$

which shows that it takes 7.8 sec. to accelerate or decelerate the velocity by 5 ft. in order not to increase or decrease the pressure to exceed 10%.

56 As may readily be noted, the above formula gives equal results for increase or decrease of pressure, and both values become infinity if any of the factors in it become infinity or zero. It is impossible, however, to imagine the pressure decrease becoming infinity; it is evident that the decrease can never exceed 100% and if this limit is surpassed, or even reached, the liquid column must sustain a rupture, and the water will flow intermittently or in the form of pulsations. From this it is evident that the formula can be correct

only, if at all, within certain limits and under certain restrictions and then only approximately, as we will see from our further discussion.

57 The above indicates that we must analyze this problem further in order to get formulae which will cover the ground and give more exact results.

58 It is a well-known fact that if penstock conditions are disturbed by moving a gate anywhere in the line, either at the upper or lower end or intermediately, the whole system becomes oscillatory, which means that the pressure and velocity in the penstock are oscillating.

59 A similar phenomenon can also be experienced in an open flume in which water is flowing. Assume that the gate at the lower end of such a flume is closed. A wave will be produced next to the gate, and this will proceed upward along the flume with a certain velocity, until it disappears either in the intake basin, where v nearly equals zero, or it will be dissipated by friction on the flume walls, depending upon the length and roughness of the flume and the velocity of the wave.

60 In closed flumes or penstocks, such waves do not disappear so quickly, the surface friction being small in comparison with the velocity and the produced force, and its influence may even be neglected.

61 It is evident that these waves, or rather vibrations, are liable to interfere with each other and consequently affect the pressure in the penstock. It is therefore of greatest importance to determine the velocity with which these vibrations travel in and along the penstock

VELOCITY OF VIBRATIONS

62 This velocity of the vibrations, which we will indicate by C in our further discussion, depends (*a*) upon the compressibility of the liquid, and (*b*) upon the nature of the material of which the penstock consists.

63 It is a well-known fact that vibrations in water proceed with the same velocity as does sound, and if the walls of the flume or penstock can be considered absolutely rigid, C therefore depending only upon the compressibility of the water, the velocity of the vibrations will be 4650 ft. per sec., or, as already stated, the same as that of sound. The penstock walls, however, are always flexible to a certain degree, depending upon the elasticity of the material from which they are constructed, and therefore they also exert a certain influence upon the vibrations.

64 Under the influence of pressure variations, the penstock walls, due to their property of elasticity, expand and contract in a rather remarkable degree, called "breathing" of penstocks. Due to this breathing, the velocity of vibration is reduced, the motion so produced naturally having a damping effect upon the vibrations, the same as any other obstruction would have. The velocity of 4650 ft. per sec. may therefore be considered a maximum with which any vibrations in the penstock will proceed.

65 The actual velocity of the vibrations can be computed by means of the following formula:

$$a = \sqrt{\frac{\frac{g}{y}}{\frac{1}{e} \times \frac{1}{E} \times \frac{D}{d}}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [14]$$

where

g = acceleration of gravity = 32.2 ft. sec.

y = specific weight of water = 62.4 lb. ft.^{3}

$\frac{1}{e}$ = elasticity of the water, e = 42,000,000 lb. ft.^{2}

$\frac{1}{E}$ = elasticity of penstock material in lb. ft.^{2}

D = diameter of penstock in ft.

d = thickness of penstock wall in ft.

66 The value of E is variable for the same material and for different materials. The following average values have been taken from Kent's Mechanical Engineer's Pocket Book and from the German engineering handbook Hütte:

For steel plates, E = 28,000,000 lb. in.^{2} = 4.032.10^9 lb. ft.^{2}

For cast iron, E = 15,000,000 lb. in.^{2} = 2.16.10^9 lb. ft.^{2}

For wooden staves, E = 1,680,000 lb. in.^{2} = 2.42.10^8 lb. ft.^{2}

67 If we substitute the numerical values for g, y, and e in formula [14] and, for the sake of brevity, place the letter $K = E^{-1} \times 10^{10}$, we will obtain

$$a = \sqrt{\frac{22{,}720}{23.5 - K \times \frac{D}{d}}} \quad \ldots\ldots\ldots\ldots\ldots [15]$$

68 The value of K can now be computed for the various penstock materials, and will be as follows:

Steel plates, K = 0.232

Cast iron, K = 0.464

Wooden staves, K = 41.50

69 Formula [14] was first derived by Lorenzo Allievi, C.E., of Rome, Italy, and published by him in 1903 in the Italian paper, Annali della Società degli Ingenieri ed Architetti, under the title Teoria generale del moto perturbato dell' acqua nei tubi in pressione. In 1904, this article was translated into French under the title Théorie générale du mouvement varié de l'eau dans les tuyaux de conduite, in the French paper Revue de Mécanique.

70 Based on the foregoing formula and values for K, Table 8 (see Appendix), giving values of a, has been computed, and includes diameters of pipes from 1 ft. to 20 ft. with thickness of walls from $\frac{1}{4}$ in. to 5 in.

71 For practical use, at least for preliminary purposes, or when the table is not at hand, the value of a can be taken as 3300 ft. per sec. for customary diameters and thicknesses of penstocks.

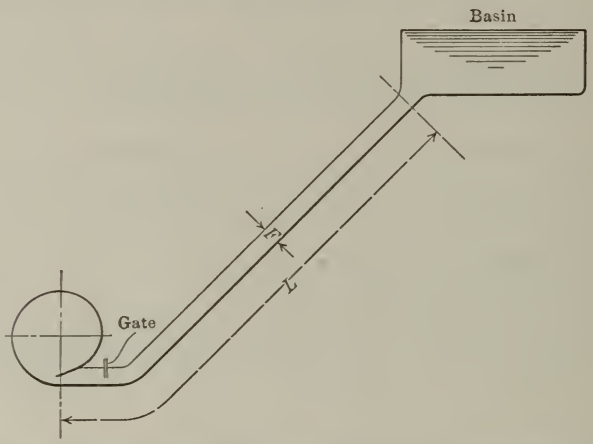


Fig. 4 Penstock Conditions resulting in Pressure Variation

72 The effect of the velocity of vibrations upon the pressure variations depends upon the regulating time T, as will be seen from the following discussion.

73 If we assume a penstock of length L, as shown in Fig. 4, with a basin at its upper end and a gate at the turbine, the water with the gate open will have a velocity v. If the gate is closed, we have a variation in the velocity, with corresponding pressure variations and vibrations. As shown above, the vibrations will travel along the pipe line with a velocity a, and will reach the basin in a time, $t = \frac{L}{a}$. If the basin is sufficiently large, that is, if its area is many times that of the penstock, then the hydro-dynamic reaction, due to the large body

of water, is the cause of a new series of vibrations which proceed down the penstock with the same velocity a, as before, and the gate is again reached by them in the time, $t = \frac{2L}{a}$.

74 These reproduced waves coming from the basin will interfere with the waves traveling upwards and have a tendency to diminish the resulting pressure variations in all sections of the penstock, therefore also those in the gate area. This would indicate the following:

a Up to the time $t = \frac{2L}{a}$ $\left(\text{period } t = 0 \text{ to } t = \frac{2L}{a}\right)$, the pressure in the gate area is constantly increasing as if the penstock were indefinitely long. (See Fig. 5.)

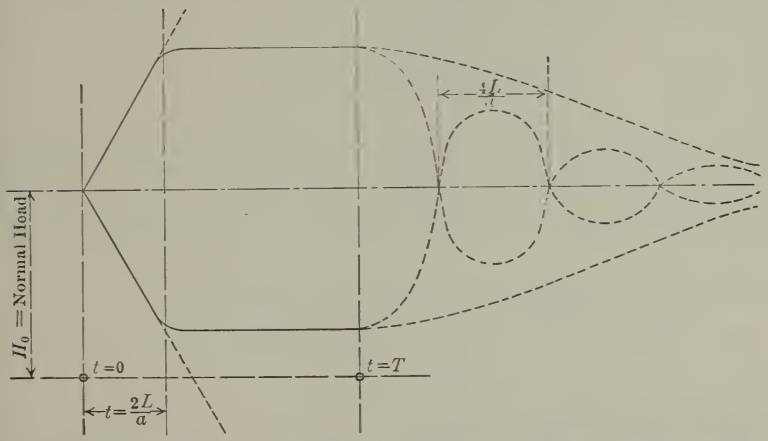


FIG. 5 EFFECT OF TIME DURING WHICH A CHANGE IN PENSTOCK CONDITIONS IS EFFECTED

b Starting from the moment $t = \frac{2L}{a}$ the pressure variations are weakened by the retroceding waves or vibrations, and we conclude that the pressure must rest constant from the time $t = \frac{2L}{a}$ until the gate stops in its travel, $(t = T)$, as both series of vibrations have an equal effect on the pressure.

c When the gate is stopped in its travel, $(t = T)$, we get different conditions depending upon whether it is fully or partially closed. If it is fully closed, fluctuations will occur having a period of $\frac{4L}{a}$, but these will become smaller and smaller until the original pressure is reached.

75 If the gate area is only partially closed in the time $t = T$, the pressure rise will decrease asymptotically, since the column of water still keeps moving, but with a changing velocity.

76 Only the period from $t = 0$ to $t = \frac{2L}{a}$ will be of interest to us in this discussion, since during this the maximum pressure rise or drop occurs.

77 The graphic illustration given in Fig. 5 explains our reasoning more clearly. From the point $t = 0$ to $t = \frac{2L}{a}$, the pressure is constantly increasing or decreasing. From the point $t = \frac{2L}{a}$ to $t = T$ the pressure remains constant.

CRITICAL TIME

78 From the above, we see at once that it would be working against desired results should we choose a regulating time T for a governor less than the value $\frac{2L}{a}$, without awaiting the weakening effect of the reflected waves which arrive at the gate in the time $\frac{2L}{a}$.

79 Therefore, the governor regulating time T should always be greater than the time $t = \frac{2L}{a}$.

80 To state this fact more clearly, we will consider both cases

$$(a) \quad T < \frac{2L}{a}$$

and

$$(b) \quad T > \frac{2L}{a}$$

81 As we have already shown, if the gate is closed in a time $< \frac{2L}{a}$, the pressure rises as long as the gate is kept moving, and if the gate is totally closed, we will obtain the maximum pressure possible in a penstock.

82 This maximum pressure can be computed by means of the formula

$$H_{max} = H_o + \frac{av}{g} \quad \ldots\ldots\ldots\ldots\ldots\ldots [16]$$

where

H_0 = initial head

v = velocity in the penstock

g = acceleration of gravity

83 This formula shows us that every change in velocity of 1 ft. means an increase or decrease in head of

$$\frac{a \times 1}{g} = \frac{3300 \times 1}{32.2}$$

$$= \text{about } 100 \text{ ft. if } T < \frac{2L}{a}$$

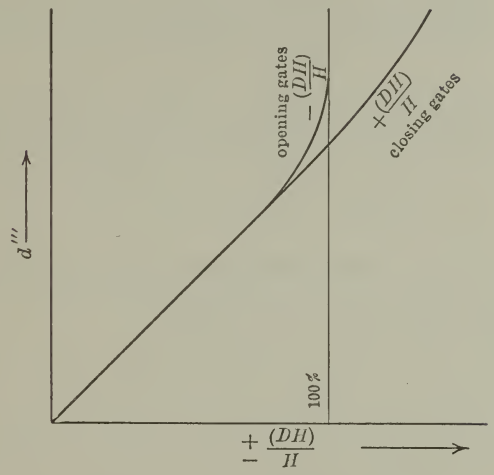


Fig. 6 Limit of Pressure Variations

84 From the example which we have considered under formula [13] we have

$$v_0 - v_1 = 5 \text{ ft.}; H = 100 \text{ ft.}; L = 500 \text{ ft.}$$

$$T = \frac{2L}{a} = \frac{2 \times 500}{3300} = 0.3 \text{ sec.}$$

$$H_{\max} = 100 + \frac{3300 \times 5}{32.2} = 600 \text{ ft.}$$

which indicates that the pressure would rise from 100 ft. to 600 ft. if we closed the gate completely in the time $t = \frac{2L}{a} = 0.3$ sec.

85 The time $t = \frac{2L}{a}$, therefore, represents a *critical time*, which

should always be calculated, and the governor regulating time chosen as much greater as other conditions will permit.

86 If we now compare formula [10], originally derived for a maximum pressure rise, with formula [16] we obtain a difference for H_{max} as follows: From formula [10]

$$\frac{(DH)}{H} = \frac{500 \times 5}{100 \times 32.2 \times 0.3} = \text{about } 2.6 = 260\%$$

$$(DH) = 260 \text{ ft.}$$

$$H_{max} = 100 + 260 = 360 \text{ ft.}$$

as compared with 600 ft. obtained from formula [16]. This clearly illustrates the importance of considering the effect of the vibrations and elasticity of the water as well as the elasticity of the penstock walls.

87 It is evident that the theory by means of which formula [16] is derived, shows the maximum value of H, as for the period

$$t = 0 \text{ to } t = \frac{2L}{a} \text{ or } \left(T \overset{=}{<} \frac{2L}{a} \right)$$

it is immaterial in what time the kinetic energy is stored in the penstock. If we close the gate completely, the same energy is stored up, whether the gate is closed in 0.1 sec. or 0.3 sec.

88 As has already been shown, for a closing time, $T > \frac{2L}{a}$, the pressure rise is constant, and this constant pressure increase or decrease represents the maximum possible value for such a regulating time, and may be calculated by means of the following formula:

$$\frac{(DH)}{H} = \frac{n}{2}\left(n \pm \sqrt{n^2 + 4}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots [17]$$

which is true for the period

$$t = \frac{2L}{a} \text{ to } t = T$$

where

$$\frac{(DH)}{H} = \text{pressure variation}$$

$$\frac{(DH)}{H} \times 100 = \text{pressure variation in per cent}$$

$$n = \frac{Lv}{gHT} = \text{formula} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [10]$$

Use + sign for closing gates and − sign for opening gates.

89 This formula [17] is most important in our calculations, since it gives us the momentary head under which a turbine must operate during a load change for any given regulating time.

90 The results of this formula plotted graphically give us two curves one for increase of head and one for decrease of head. For an increase, if n = infinity, $\frac{(DH)}{H}$ = infinity; for a decrease, if n = infinity, $\frac{(DH)}{H}$ will asymptotically reach the value 1. (See Fig. 6.)

91 It will be noted that formula [17] contains just those factors which we missed in formula [10]. According to formula [12], our results, if plotted, would appear along the straight line as shown, practically midway between the two curves, $+\frac{(DH)}{H}$ and $-\frac{(DH)}{H}$. (See Fig. 7.)

92 We note further that for a regulating time greater than $\frac{2L}{a}$ and variations of pressure not exceeding 20 to 25%, formula [10] may be used approximately. For larger variations, the difference between $+\frac{(DH)}{H}$ and $-\frac{(DH)}{H}$ becomes considerable, and formula [17] must be used even for approximate results.

93 Tables 3 to 8 (see appendix) have been computed from formula [17] and are based on different values of $\frac{L}{H}$, (where L is the length of the penstock and H the effective head), and for velocities up to 20 ft., each table giving the pressure variations both $+\frac{(DH)}{H}$ and $-\frac{(DH)}{H}$ for a given regulating time T.

94 These tables will be found of great benefit in determining quickly the regulating time and the corresponding pressure variations for different penstock diameters, provided L and H are given, as they are in practically every case. But in determining or assuming the regulating time it must be remembered that it must always be greater than $\frac{2L}{a}$.

95 Referring to the tables and the case already used to illustrate, namely, L = 500 ft., H = 100 ft., v = 5 ft. and T = 2 sec.

$$\frac{L}{H} = \frac{500}{100} = 5$$

$$\frac{(DH)}{H} = +0.471 \text{ and } -0.32 = 47.1 \text{ and } 32 \text{ per cent}$$

From formula [10] we would obtain

$$\frac{(DH)}{H} = 39 \text{ per cent}$$

as the approximate average value of the exact results. The tables permit of interpolation.

96 It now becomes necessary to analyze the effect of the pressure variations on the speed regulation.

97 As already stated, if the regulating time T is chosen greater than $\frac{2L}{a}$ the pressure rise or drop, and therefore the head remains constant and its maximum value may be calculated from formula [17], according to which the tabular values are found. We may, therefore, assume that the turbine is operated at a constant head during any load change, the head varying for different per cent of changes; and the head for any part load change may be found by substituting the new penstock velocity in formula [17], or directly from the tables.

98 From formula [3] we see that d is directly proportional to N (= h.p.), the load. Therefore, if the head is varied, the capacity varies as $\sqrt{H^3}$ (h.p. varies as the square root of the head cubed), and for a new head = $\Big(H + (DH)\Big)$ the capacity of the turbine rises with $\Big(H+(DH)\Big)^{\frac{3}{2}}$ and the new capacity of the turbine is $N\Big(H + (DH)\Big)^{\frac{3}{2}}$, which is the case when load is thrown off.

99 When the load is thrown on the available energy decreases with the term

$$\frac{1}{\Big(H - (DH)\Big)^{\frac{3}{2}}}$$

100 To obtain the results in per cent, these terms are transformed as follows: For closing gates

$$\left(1 + \frac{(DH)}{H}\right)^{\frac{3}{2}}$$

and for opening gates

$$\frac{1}{\left(1 - \frac{(DH)}{H}\right)^{\frac{3}{2}}}$$

so that finally we get a speed variation d modified by pressure variations $\pm \frac{(DH)}{H}$ as given by the following formulae: For closing gates (load thrown off)

$$d''' = d_{\text{eff}} \left(1 + \frac{(DH)}{H}\right)^{\frac{3}{2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [18]$$

for opening gates (load thrown on)

$$d''' = \frac{d_{\text{eff}}}{\left(1 - \frac{(DH)}{H}\right)^{\frac{3}{2}}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [19]$$

101 After making a number of calculations according to these formulae, it will be noted that the results from both are practically identical within certain limits. That they are not identical throughout the range of the pressure variation tables can be seen from the following: For

$$-\frac{(DH)}{H} = 1 \ (100\ \%)$$

$$d''' = \frac{d_{\text{eff}}}{(1-1)^{\frac{3}{2}}} = \text{infinity}$$

whereas for

$$+\frac{(DH)}{H} = 1$$

$$d''' = d_{\text{eff}}\,(1+1)^{\frac{3}{2}} = \text{a definite value}$$

102 This shows us that up to a point where $\frac{(DH)}{H} = \pm 1$, the results for d''' are practically identical for opening or closing the gates. Then the d''' curve turns up for opening and becomes infinity with $-\frac{(DH)}{H} = 1$, while the d''' curve for closing gates goes on undisturbed as shown by the accompanying diagram (Fig. 6).

103 This indicates that for practical problems, where $-\frac{(DH)}{H}$ would never be allowed to reach the value of 1, it is not necessary to calculate d''' for both opening and closing gates, but that either one of the formulae [18] and [19] will give the desired results. As $-\frac{(DH)}{H}$ approaches 100%, d''' for opening gates becomes considerably greater than d''' for closing gates. In such cases, it is well to calculate d'''

for both movements, particularly as frequently it is not necessary that d''' for opening gates be as low as d''' for closing gates.

104 From observation of the tables on pressure variations, it will be noted that a change in velocity of 1 ft. will have considerable effect on the pressure variation, and this velocity should always be kept as low as is consistent with a reasonable investment. We should always bear in mind that by decreasing our penstock velocities, we not only help our pressure variations, and hence speed regulation, but use the water more economically, since our friction head will be less as the velocity becomes less.

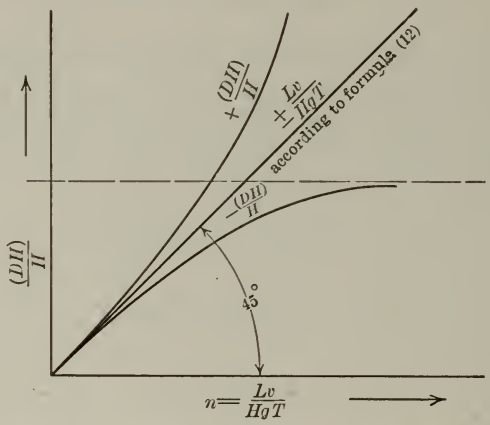


FIG. 7 COMPARISON OF PRESSURE VARIATIONS FOR RIGID AND FLEXIBLE PENSTOCKS

PRESSURE REGULATOR

105 If a reasonable solution cannot be found in the foregoing, such as will satisfy both engineering and commercial conditions, or either, it will become necessary to consider one or more practical appurtenances which are discussed in what follows. Economically it will often prove desirable to consider these, even if not necessary for a solution of the problem from an engineering point of view.

106 A pressure regulator is a device attached to the turbine casing or penstock near the turbine, and operated from the governor in such a manner that when the turbine gates are closed suddenly and a sufficient amount to disturb the regulation on account of pressure rise, the regulator will be opened by the governor, sufficiently to keep the pres-

sure rise within reasonable limits. Such a pressure regulator should be adjustable so that the amount of pressure rise for any given load change can be predetermined for any set of conditions. It should close automatically and slowly under usual conditions so that its sudden closure will not produce a pressure rise. It should, however, be so arranged, that if a sudden increase should occur after a sudden decrease in load during which the regulator is opened, the governor will close the regulator quickly and positively. Otherwise water then needed to prevent an excessive pressure drop would pass through the regulator while it was slowly closing. Such pressure regulators can be designed practically, with all the functions described, and if of sufficient size the maximum pressure rise will not exceed 10% above normal with a load change of 100%. It will be seen from our subsequent discussion that for a fixed set of conditions, the pressure drop is always less than the pressure rise, which at once shows us the value of the pressure regulator.

107 From tests made by the writer and substantiated by others, the pressure rise in a turbine, and consequently in the penstock feeding a turbine, depends directly upon the discharge capacity of the pressure regulator, if properly designed, and upon the sensitiveness of the operating mechanism of the pressure regulator. The latter should be adjustable, so that the pressure regulator can be set to meet the requirements of any fixed conditions after being installed.

108 If the sensitiveness of the pressure regulator is 10%, that is, if the operating mechanism is so adjusted that it will open if a sudden load change amounting to 10% of the total load occurs, the pressure rise will be as follows:

Discharge capacity, 100% of turbine discharge, pressure rise 10%
Discharge capacity, 75% of turbine discharge, pressure rise 20%
Discharge capacity, 50% of turbine discharge, pressure rise 30%

109 Due to the insensitiveness of the operating mechanism, the pressure rise will be the same regardless of the load, and will be even slightly greater for a small load change than a large one, as it will take proportionally more time to open sufficiently to pass the small quantity of water. Another factor which helps to bring this about is the discharge coefficient of the pressure regulator, as this increases with the size of the opening. In calculating the effect on the speed regulation of the pressure rise, it is therefore necessary to take this into consideration for a partial load change as well as a full load change.

110 As already explained, although $-\frac{(DH)}{H}$ is always less than $+\frac{(DH)}{H}$, the corresponding regulation as affected by these pressure variations is practically the same, at least within the usual limits. If this is the case, it would appear that a pressure regulator is of no value in improving speed regulation. However, in most plants the load is thrown off much more suddenly than it is thrown on, the few exceptions usually occurring in small plants which always require more careful consideration with respect to the subject of speed regulation.

111 A lighting load is usually turned on as suddenly as off, but its proportion to the total load is exceedingly small, a few lights ordinarily being turned on at a time. Motor loads are usually turned on very slowly by means of resistances or otherwise, and such sudden loads as may come on the motor during its operation are generally only a small part of the total load. On the other hand, a motor load is usually thrown off suddenly, whether purposely or accidentally, so that it becomes more difficult to maintain constant speed for load thrown off than for load thrown on, and it is of great benefit to have conditions for good regulation better for that case.

112 This benefit is obtained by means of the pressure regulator, as $+\frac{(DH)}{H}$ may be reduced to such an extent that its effect on the speed regulation is very small.

113 It is of course evident that a pressure drop of less than 100% is not dangerous to the penstock, whereas a pressure rise much less than 100% may be very dangerous, particularly if it occurs frequently. In this case it is liable to crystallize the penstock material, especially at the rivetted joints, so that the penstock may give way with a pressure rise much within the limit set by the original factor of safety. In such a case, the pressure regulator not only assists in the regulation of speed but protects the penstock as well.

114 When a pipe line or penstock is laid over a rough country, or where it descends precipitous hillsides after running along a comparatively gentle slope, care must be taken that the water will never be accelerated so quickly at a change in slope that the water column can break and thus exceed or even approach the maximum pressure drop of 100%. In such a case the penstock is in danger either of collapsing or of being shattered by a possible water ram.

115 It is uncommercial to provide against something that may never happen and if the greatest load change ever expected under normal operation is 25%, it would be a waste of money to provide for 100% change of load, as far as speed regulation is concerned.

116 Yet it is always necessary to consider the effect of the maximum pressure rise for 100% load change, since in the case of a short circuit or break in the governor such an extreme load change may occur and subject the penstock to the corresponding pressure rise.

117 It may therefore be desirable to install a pressure regulator simply to protect the penstock with entire disregard of the regulation, which may be commercially satisfactory without it.

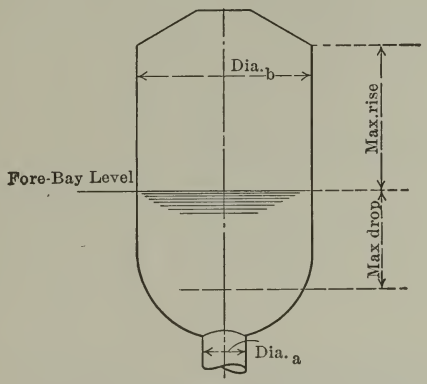


FIG. 8 PRACTICAL DESIGN FOR A HIGH STANDPIPE

EQUALIZING RESERVOIR AND STANDPIPE

118 As has just been stated above, the pressure drop is always less than the pressure rise, and as long as the pressure drop is within such limits as will permit satisfactory regulation, the pressure rise can be kept within safe limits by means of the pressure regulator. When, however, the point is reached where the pressure drop is excessive, the problem is again altered. The practical solution of such a problem is by means of an "equalizing reservoir," or a "standpipe." The effect of either is the same, namely, to reduce the effective length of the penstock, and to supply or take up water during a change of load, while the flow of the same amount of water will be accelerated or retarded in that portion of the penstock beyond the reservoir or standpipe. In small plants the equalizing reservoir is seldom used, as the

same effect can be had more cheaply with a standpipe. However, in large plants under medium heads, where the quantity of water used is considerable, the reservoir will usually prove more economical in first cost and of much greater benefit to pressure variations and hence to speed regulation. The larger the surface area of a reservoir or standpipe, the smaller will be the pressure variation, which at once shows the advantage of the reservoir.

119 The topography of the country surrounding a power site usually decides whether a reservoir or standpipe should be used, as entirely artificial reservoirs may prove very expensive, and in such cases the standpipe is resorted to. It will often be found advantageous to change the shortest or most economical route of the pipe line to one more circuitous, in order to use a natural reservoir site, or to bring

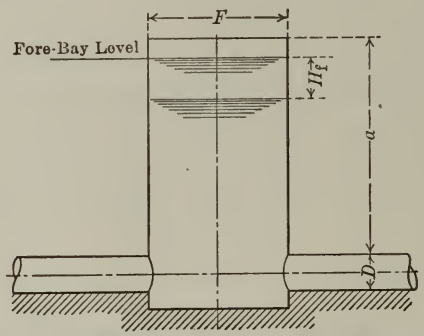


FIG. 9 MINIMUM DIMENSIONS OF A STANDPIPE

either the reservoir or the standpipe closer to the power house. If the standpipe is of suitable diameter and close to the turbine, the speed regulation will approach that obtainable with an open flume. Otherwise the problem becomes that of a plant with a closed penstock, of a length equal to that of the draft tube, plus the length of the penstock from the turbine to the standpipe, plus the height of the standpipe itself. To approach more nearly the effect of regulating reservoirs, high standpipes should have their upper part enlarged, in the shape of a tank (Fig. 8). This tank may be supported on structural steel columns. The pipe leading to this tank should be of a diameter not less than that of the penstock. Where a power house is located near a gently sloping hillside, the standpipe may be laid up this hillside and supported by it, instead of being supported by columns or

otherwise. Standpipes for heads of a thousand feet have thus been built.

120 The standpipe should be located as near the turbine as possible, as has been previously stated. If it is arranged with an overflow, the pressure rises can be practically eliminated, and the pressure drop will depend directly upon the size or capacity.

121 The minimum height of such a standpipe is determined by the restriction that in no case must the water level in it drop to such a point as would admit air into the penstock. This condition is satisfied by the following equations (Fig. 9):

$$a > \sqrt{\frac{Q}{F} \times \frac{L \times v}{g}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [20]$$

$$a \geqq D + H_f \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [21]$$

where

Q = cu. ft. per sec.
F = area of standpipe in sq. ft.
L = pipe line length above standpipe in ft.
v = pipe line velocity in ft. per sec.
H_f = friction head in ft.
D = diameter of penstock in ft.

122 It is seldom satisfactory to have the standpipe overflow and furthermore it is not economical. It is usually built high enough so that the water cannot overflow, even with a maximum load change. In northern climates, where there is danger of freezing, the entire standpipe should be well lagged and quite often it must be provided with steam-pipe coils, supplied with steam from a boiler installed at the foot of the standpipe.

123 In order to reduce the height of the standpipe as much as possible, the slope of the pipe line should be as slight as is consistent with the velocity head and friction head required, as the top of the standpipe must in any case be higher than the level of the water in the forebay to meet the conditions of a shutdown with pipe line full.

124 If load is suddenly thrown on, a certain time will elapse before the water accelerates in that part of the pipe between the standpipe and the turbine, and another and longer time will elapse before the water in the part of the pipe above the standpipe accelerates to the velocity required by the new load. During the latter time, the standpipe must supply the turbine with a quantity of water which is the difference between that used by the turbine and that supplied

by that part of the pipe above the standpipe at the reduced velocity during acceleration. The reverse is true when load is thrown off.

125 The actions of the water in a plant with standpipe or equalizing reservoir are easily explained, but very difficult to compute. In the following will be found a practical method for calculating standpipe dimensions. No attempt is made to give the derivation of the formulae, as that in itself would require a paper of some length.

126 Let

L = length of pipe line between standpipe and intake
D = average diameter of pipe line in ft.
F_p = average area of pipe line in sq. ft.
F_s = average area of standpipe in sq. ft.
Q = water discharged by turbines in cu. ft. per sec.
v = average velocity of water in pipe line in ft. per sec.
y = drop of water level in standpipe when Q cu. ft. of water is flowing continuously
y_{max} = maximum drop or rise of water level
h_f = friction loss in ft. in the pipe line $= f\frac{L}{D}\frac{v^2}{2g} = av^2$
$a = \frac{fL}{D(2g)}$ = a function of the velocity, however assumed to be constant
f = friction coefficient = 0.015 for new plate steel pipe
g = 32.2 ft. per sec. = acceleration of gravity
Dy = change of level of standpipe
Dv = change of velocity in pipe line
Dt = time element in sec.

Then

$$Dy = \left(\frac{Q}{F_s} - \frac{F_p}{F_s}v\right)Dt \quad \text{.....[22]}$$

and

$$Dv = \frac{g}{L}\left(y - av^2\right)Dt \quad \text{.....[23]}$$

127 From these two formulae y can be computed for any time interval and the results plotted with time in seconds as abscissae and y in feet as ordinates.

128 The resulting curve (Fig. 10) shows the maximum drop or rise of the water level for any assumed or existing pipe line and standpipe combination.

129 If it is desired to approximate the proper capacity of a standpipe or equalizing reservoir in order not to exceed an assumed or desired pressure rise or drop, we must proceed as follows.

130 Referring to Par. 54 we find the formula for the accelerating or decelerating time for a column of water as follows:

$$T = \pm \frac{h\,(Dv)}{Hg\,\dfrac{(DH)}{H}}$$

$\dfrac{(DH)}{H}$ is the assumed or desired pressure rise or drop.

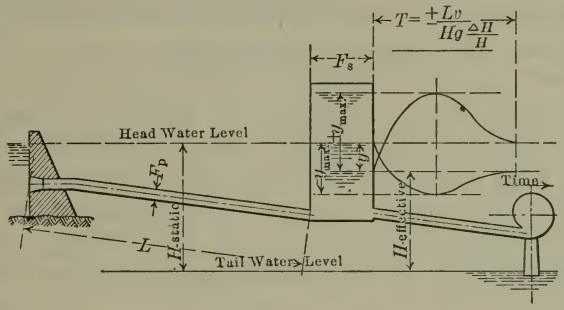


Fig. 10 Pressure Variations with Standpipe

131 If we now let S represent the governor opening or closing time, then the equalizing capacity $C = Q\ (T - S)$ in cu. ft. and the area

$$F_s = \frac{C}{h} \qquad [24]$$

where

$$h = h_f + \frac{(DH)}{H}\left(H \pm h_f\right) \qquad [25]$$

132 The theory from which these formulae result was first derived by Engineer Pressel and published by him in the Schweizerische Bauzeitung, January 1909, under the title Dimensionierung von Wasserschloessern.

133 A very thorough, though highly theoretical study of this subject was published in the same journal in 1908 by Prof. Franz Prasil, under the title of Wasserschlossprobleme.

134 Although not so economical in the use of water, it will often prove more economical in first cost of plant to install a pressure regulator, rather than build the standpipe to the height as found necessary by the preceding equations. With a suitable pressure regulator in the system, the height of the standpipe above the forebay water level is more or less a question of judgment, but if made equal to the depth, it will always prove sufficient. It should be remembered, however, that a pressure regulator is only a mechanical device, somewhat complicated, and many engineers would be unwilling to risk a catastrophe, in case of a sudden full load change with the pressure regulator out of order.

135 With the same purpose in view, namely, the reduction of the height of the standpipe, it may sometimes prove expedient to provide water rheostats connected to the low-tension bus-bars, with an oil switch in the connection. By means of a mechanical trip on the governor or gate mechanism the control circuit of the switch can be closed at any desired gate opening or power output, so that in case of a short circuit part of the load will be switched onto the water rheostats. This will reduce the pressure rise and therefore the height of the standpipe necessary and will at the same time keep the voltage of the generators in perfect control. After the load is again picked up, the switches connecting the rheostats can be opened from the switchboard, the rheostats being left in the water at all times. This scheme is of course applicable only under certain load conditions.

136 Equalizing reservoirs are usually provided with an overflow or spill-way, so that their maximum height need not exceed that of the forebay level.

137 From the foregoing it will be seen that the location of the standpipe or regulating reservoir cannot always be as desired, and the solution of the problem of pressure variation is in many cases only partially solved by them. It is true that an entirely artificial reservoir or a standpipe could be located almost at will, but the cost will usually prove excessive, and, unless a natural site for either is available, their distance from the power house will reduce itself largely to a question of cost, and the cost of increased flywheel effect must be balanced against the cost of closer proximity of standpipe to turbine, to produce the desired regulation.

138 As already stated in our discussion of the pressure regulator, the pressure drop is always less than the pressure rise, so that it is necessary only to bring the standpipe near enough to prevent excessive pressure drop, disregarding the pressure rise and providing for it by means of a pressure regulator.

139 The capacity of a standpipe must be the same regardless of the distance it is located from the turbine, provided the other conditions, such as maximum change of load to be provided for and regulalating time are the same. If a greater regulating time is allowed on account of having a longer penstock, the capacity of the standpipe could theoretically be reduced slightly, but this should not be done in practice, and the formulae given therefore hold good for any conditions.

140 In a plant with long penstocks, where it is impossible to install a standpipe, and where it is practically out of the question to increase the size of the penstocks on account of excessive cost or otherwise, it will often be found impracticable to provide sufficient flywheel effect to obtain satisfactory regulation.

SYNCHRONOUS BYPASS

141 If this is the case in connection with reaction turbines, a synchronous bypass must be provided which should be exactly what the name implies; that is, it should discharge that part of the full load flow of the water which is not passing through the turbine at any given time. If the flow of the water through the turbine is changed in volume, the change in flow through the bypass should correspond.

142 It is very difficult to design a bypass that will do this, since the discharge through the turbine gates is not proportional to the governor stroke, from which the bypass must be operated, and furthermore the coefficient of discharge of the bypass cannot be kept constant during its full range of opening. Due to these causes and the insensitiveness of the governor, the pressure variations both ways may be quite considerable with the synchronous bypass and should be taken into account in computing the speed regulation. They may be as high as 25%, but the writer's experience shows that they can be kept as low as 10% for both opening and closing gates. The bypass should always be brought as near as possible to the turbine gates and in line with the flow of water into them.

DEFLECTING NOZZLE

143 In the case of impulse turbines, the synchronous bypass may also be used, but the deflecting nozzle is more generally used in practice, as it has several important advantages. With the latter the flow of water is not interfered with, but the stream is partially or wholly

deflected from the buckets, maintaining practically constant pressure, and varying the active volume only. The problem therefore reduces itself to one identical with that of a turbine in open flume, or without pressure variations, and the regulation is subject only to the fly-wheel effect.

144 Both of the devices mentioned above, which are too well known to require detailed description, are extremely wasteful, and although numerous modifications have been made to reduce their wastefulness, such devices usually interfere with automatic regulation and the lessening of their wastefulness is one of degree only. In places where fixed quantities of water must be passed, regardless of the load on the plant, such as in plants where no storage whatever is available, or where irrigation or other laws require it, these two methods of regulation may prove the proper solution.

145 No solution of a problem in speed regulation is commercially correct, unless accompanied by an analysis of the probable load curve of the power plant to be regulated. It is evidently useless to provide against something which will never occur. Thus, if a plant consists of a number of large units, the greatest load change to be considered should be carefully gone into. This decided, it should be known over how many units the load change must be distributed; in other words, what will be the least number of units running in parallel on one circuit. In a large plant it is very seldom necessary to consider a total load change (100% change of load), whereas in small plants this may frequently occur, especially when only one unit is running.

AIR CHAMBER

146 Upon first consideration, it appears that an air chamber would be an ideal means of providing against extreme pressure variations. The necessary volume of such a device can be computed from the following formula:

$$V = \frac{T \times v}{\frac{(D\,H)}{H}} \qquad [26]$$

where

V = volume of air chamber in cu. ft.

T = accelerating time $= \frac{Lv}{gH}$

v = velocity in penstock in ft. per sec.

$\frac{(DH)}{H}$ = permissible pressure variation

147 It will usually be found that extremely large and therefore expensive air chambers are required. However, leaving the cost out of consideration, there is a constant danger connected with this device, since the water absorbs the air, so that it may not only be of no use, but may greatly increase the pressure variations. Although such modifications of the simple air chamber as automatically operated air compressors and check valves at the penstock connection have been suggested, they should be used only as a last resort.

ADDENDUM

148 Since this paper was written, a hydro-electric plant has been built and put in operation on the White River in Washington, which embodies almost every device described in the paper to secure good commercial regulation.[1]

149 The hydraulic power units installed are the largest ever put in operation and they are supplied with water from a closed conduit consisting of tunnel and penstocks with a total length of about 5000 ft. The tunnel is 12 ft. in diameter and about 2800 ft. long. It carries the water for all the turbines. The penstocks are each about 2200 ft. long and vary in diameter, being 8 ft. in diameter at the upper end and 6 ft. in diameter at the lower end. The average normal penstock velocity is about 9 ft. per second and the maximum about 11 ft. per second.

150 The provisions made to secure safe operation with resulting commercial regulation consist of (*a*) standpipe or open forebay, 30 ft. in diameter and 35 ft. high, with spillway in tunnel; (*b*) steel standpipe on the brow of the hill over which the penstock leads to the turbine below, each penstock having its separate standpipe 6 ft. in diameter and 85 ft. high, with top extending 25 ft. above high water; (*c*) each penstock is provided with two air chambers just outside of the power house, both being 7 ft. in diameter and 75 ft. high, and containing 2000 cu. ft. of air under 190 lb. pressure maintained by the hydraulic head; (*d*) each turbine is provided with a 30-in. governor operated pressure regulator; and (*e*) each turbine is provided with a 30-in. pressure operated relief valve.

151 In addition to the above mentioned devices, which are all safety devices as well as regulating devices, bursting plates have been inserted in a separate discharge pipe for additional safety.

[1] Engineering News, April 11, 1912.

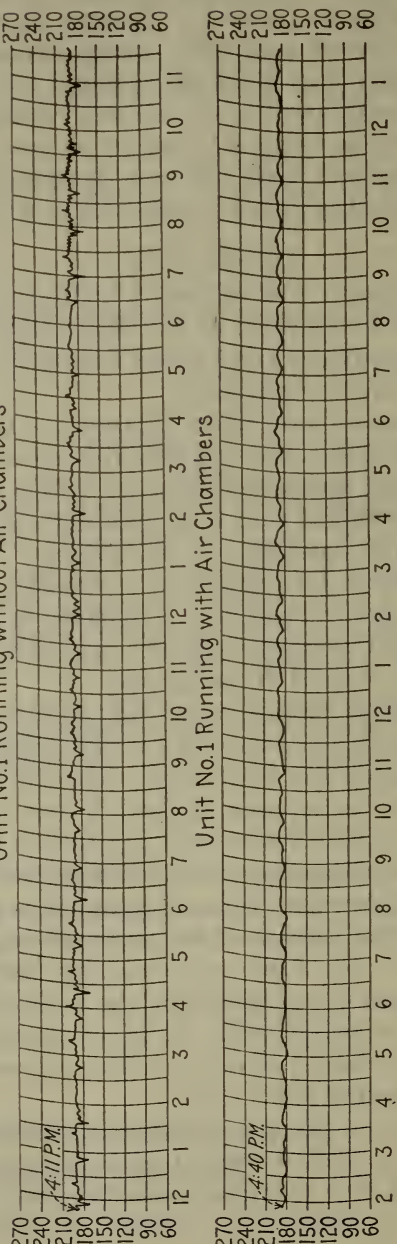


FIG. 11 PRESSURE CURVES

152 To the knowledge of the writer this is the first plant in which large enough air chambers have been installed, with all provisions necessary to insure them against becoming useless or harmful.

153 The air cushion tanks are fitted with floats and valves to allow free water passage in and out, but to prevent escape of air into the penstocks.

154 A 550-cu. ft. motor-driven air compressor is located in the power station and connected to the air chambers, so that any leakage of air may be provided for, and the proper amount of cushion be always insured.

155 The accompanying pressure curves, Fig. 11, show how the pressure variations are effected during commercial operation. These are presented through the courtesy of Mr. Shuffleton of the Stone & Webster Engineering Corporation. It will be noted that without the air chambers in commission the pressure varies through a range of about 35 lb. maximum, whereas with the air chambers this variation is reduced to about 10 lb.

156 It is evident however that the latter variation is distributed over such an extended time interval that it will have practically no effect on the speed regulation, whereas without the pressure chambers, the variations are sudden and very detrimental to speed regulation.

157 In this plant these air chambers are therefore of unquestionable benefit to speed regulation, but it is evident that in order to get good results, such chambers must be designed to have liberal dimensions.

APPENDIX

TABLE 1 CHARACTERISTICS OF STANDARD REACTION TURBINE RUNNERS

Specific speed $k = \frac{\text{r.p.m.}}{H}\sqrt{\frac{\text{h.p.}}{\sqrt{H}}}$ = speed in r.p.m. of a turbine runner diminished in all dimensions to such an extent as to develop 1 h.p. when working under the head H = 1 ft.

h.p.$_1$ = horsepower per 1-ft. head = $\frac{\text{h.p. for any head}}{H. \times \sqrt{H}}$. Power varies as $\sqrt{H^3}$. Therefore h.p. for any head = h.p.$_1 \times H \times \sqrt{H}$.

r.p.m.$_1$ = revolutions per 1-ft. head = $\frac{\text{r.p.m.}}{\sqrt{H}}$. Speed varies as $\sqrt{H}$. Therefore r.p.m. = r.p.m.$_1 \times \sqrt{H}$.

Q_1 = quantity per 1-ft. head = $\frac{Q}{\sqrt{H}}$. Discharge varies as $\sqrt{H}$. Therefore $Q = Q_1 \times \sqrt{H}$

Type *A*				Type *B*			Type *C*			Type *D*		
Specific speed k = 13.55				k = 20.3			k = 29.4			k = 40.7		
Peripheral speed u_1 = 0.585				u_1 = 0.625			u_1 = 0.665			u_1 = 0.70		
Diam.	r.p.m.$_1$	h.p.$_1$	Q_1	r.p.m.$_1$	h.p.$_1$	Q_1	r.p.m.$_1$	h.p.$_1$	Q_1	r.p.m.$_1$	h.p.$_1$	Q_1
15	71.7	0.0358	0.38	76.7	0.0705	0.791	81.5	0.130	1.35	85.6	0.226	2.38
18	59.8	0.0514	0.55	64.0	0.105	1.10	68.0	0.187	1.95	71.5	0.324	3.42
21	51.2	0.0705	0.75	54.7	0.1385	1.45	58.2	0.225	2.34	61.3	0.442	4.64
24	44.75	0.0915	0.975	47.8	0.182	1.91	51.0	0.333	3.46	53.55	0.577	6.08
27	39.8	0.116	1.235	42.5	0.229	2.41	45.2	0.423	4.40	47.6	0.731	7.70
30	35.8	0.1425	1.52	38.25	0.284	2.982	40.2	0.520	5.40	42.8	0.902	9.47
34	31.65	0.184	1.96	33.8	0.363	3.82	36.0	0.668	6.95	37.8	1.158	12.20
38	28.25	0.23	2.442	30.2	0.453	4.78	32.2	0.835	8.68	33.8	1.444	15.20
42	25.65	0.28	2.98	27.4	0.551	5.80	29.1	1.016	10.55	30.6	1.765	18.50
46	23.40	0.336	3.58	25.0	0.665	7.00	26.6	1.225	12.78	27.9	2.120	22.3
50	21.50	0.398	4.24	22.9	0.790	8.40	24.4	1.450	15.10	25.65	2.50	26.3
55	19.56	0.48	5.12	20.9	0.950	10.00	22.25	1.745	18.10	23.30	3.04	32.0
60	17.90	0.573	6.10	19.1	1.130	11.90	20.4	2.08	21.60	21.40	3.61	38.0
65	16.55	0.672	7.15	17.65	1.330	14.00	18.8	2.44	25.40	19.80	4.22	44.2
70	15.40	0.785	8.35	16.40	1.535	16.15	17.5	2.82	29.40	18.35	4.90	51.5

TABLE 1—Continued

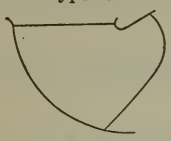
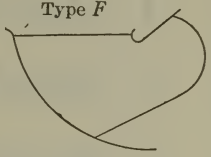
Type *E*				Type *F*			
							
k = 49.7 — 62.8				k = 70.0 — 83.85			
u_1 = 0.72 — 0.78				u_1 = 0.835 — 0.89			
Diam.	r.p.m.$_1$	h.p.$_1$	Q_1	r.p.m.$_1$	h.p.$_1$	Q_1	Diam.
14	94.6	0.277	2.95	110.9	0.410	4.53	14
16	83.6	0.367	3.90	97.0	0.541	5.96	16
18	74.3	0.471	5.02	86.8	0.704	7.77	18
20	67.0	0.597	6.37	77.0	0.912	10.01	20
22	61.0	0.731	7.80	69.7	1.133	12.55	22
24	56.0	0.883	9.42	64.3	1.375	15.20	24
26	52.0	1.055	11.20	59.5	1.623	17.95	26
28	48.1	1.243	13.25	55.3	1.930	21.30	28
30	45.5	1.436	15.30	52.0	2.20	24.30	30
32	42.7	1.65	17.55	49.1	2.555	28.25	32
34	40.3	1.89	20.10	46.6	2.825	31.20	34
36	38.2	2.15	22.80	44.1	3.14	34.75	36
38	36.2	2.42	25.70	41.9	3.52	38.90	38
40	34.5	2.75	29.3	39.8	3.93	43.40	40
42.5	32.5	3.09	33.0	37.5	4.33	47.80	42.5
45	30.8	3.53	37.6	35.8	4.92	54.25	45
47.5	28.8	4.01	42.65	34.0	5.66	62.60	47.5
50	27.8	4.45	47.4	32.3	6.13	67.80	50
52.5	26.6	4.95	52.7	30.8	6.75	74.60	52.5
55	25.45	5.52	58.8	29.9	7.50	82.75	55
57.5	24.3	6.10	65.0	28.2	8.16	90.00	57.5
60	23.4	6.80	72.4	27.05	8.94	98.70	60
64	22.1	7.63	81.25	25.60	10.05	111.0	64
68	20.9	8.57	91.3	24.20	11.34	125.0	68
72	19.8	9.58	102.0	22.58	13.07	144.0	72
76	18.8	10.92	116.0	21.50	14.52	161.0	76
80	17.9	12.30	131.0	20.36	16.29	180.0	80
84	....	....	.....	19.50	17.88	197.0	84
88	....	....	.....	18.60	20.4	225.0	88
92	....	....	.....	17.75	22.0	243.0	92
96	....	....	.....	16.85	24.48	270.0	96
100	....	....	.....	16.30	26.46	293.0	100

TABLE 2 PRESSURE VARIATIONS FOR OPENING OR CLOSING TIME OF 1 SEC.

+ Sign Indicates Pressure Rise; — Sign Pressure Drop. To Get Results in Per Cent Multiply Tabular Values by 100.

$\frac{L}{H}$		Velocities in Penstock in Ft. per Sec.														
		1 ft.	2 ft.	3 ft.	4 ft.	5 ft.	6 ft.	7 ft.	8 ft.	9 ft.	10 ft.	12 ft.	14 ft.	16 ft.	18 ft.	20 ft.
1	+	0.032	0.064	0.098	0.132	0.169	0.205	0.243	0.283	0.322	0.372	0.450	0.545	0.640	0.740	0.850
	−	0.032	0.060	0.090	0.116	0.144	0.172	0.195	0.220	0.244	0.274	0.311	0.351	0.390	0.425	0.460
1.5	+	0.047	0.096	0.150	0.205	0.260	0.322	0.386	0.446	0.520	0.587	0.74	0.90	1.08	1.26	1.46
	−	0.045	0.088	0.131	0.169	0.206	0.244	0.279	0.309	0.342	0.391	0.425	0.475	0.520	0.560	0.595
2	+	0.065	0.132	0.207	0.280	0.360	0.450	0.543	0.638	0.740	0.840	1.07	1.33	1.62	1.91	2.22
	−	0.060	0.116	0.171	0.219	0.265	0.311	0.351	0.388	0.425	0.456	0.515	0.575	0.618	0.655	0.69
2.5	+	0.081	0.168	0.270	0.361	0.470	0.590	0.716	0.850	0.990	1.14	1.46	1.84	2.25	2.69	3.19
	−	0.075	0.143	0.215	0.265	0.320	0.370	0.415	0.460	0.497	0.534	0.593	0.642	0.691	0.727	0.758
3	+	0.098	0.205	0.322	0.450	0.588	0.740	0.905	1.08	1.26	1.47	1.91	2.42	2.99	3.60	4.31
	−	0.090	0.169	0.244	0.310	0.368	0.425	0.471	0.518	0.560	0.598	0.655	0.705	0.748	0.780	0.815
3.5	+	0.116	0.243	0.385	0.544	0.708	0.90	1.11	1.33	1.55	1.86	2.41	3.06	3.81	4.66	5.60
	−	0.104	0.193	0.276	0.550	0.415	0.475	0.528	0.580	0.612	0.655	0.707	0.752	0.791	0.822	0.85
4	+	0.132	0.284	0.453	0.635	0.840	1.07	1.33	1.62	1.91	2.26	2.97	3.82	4.80	5.86	7.10
	−	0.116	0.220	0.312	0.386	0.456	0.522	0.572	0.615	0.655	0.693	0.752	0.791	0.82	0.850	0.89
4.5	+	0.150	0.323	0.520	0.736	0.985	1.26	1.57	1.910	2.28	2.69	3.60	4.66	5.87	7.22	8.72
	−	0.130	0.244	0.343	0.425	0.495	0.556	0.612	0.655	0.692	0.728	0.780	0.822	0.85	0.882	0.90
5	+	0.168	0.364	0.592	0.848	1.130	1.46	1.86	2.24	2.69	3.19	4.26	5.60	7.12	8.75	10.6
	−	0.144	0.266	0.371	0.460	0.531	0.595	0.654	0.693	0.730	0.758	0.810	0.840	0.885	0.910	0.915
6	+	0.206	0.450	0.739	1.08	1.46	1.90	2.44	2.99	3.60	4.31	5.88	7.75	9.90	12.2	14.9
	−	0.171	0.310	0.425	0.515	0.590	0.652	0.712	0.758	0.782	0.812	0.862	0.892	0.917	0.94	0.94
7	+	0.243	0.540	0.902	1.32	1.82	2.40	3.06	3.82	4.66	5.60	7.70	10.2	13.0	15.3	19.9
	−	0.195	0.350	0.478	0.570	0.640	0.712	0.746	0.792	0.822	0.84	0.89	0.92	0.93	0.96	0.96
8	+	0.285	0.640	1.08	1.62	2.24	2.97	3.84	4.81	5.88	7.12	9.80	13.1	16.8	21.0	25.8
	−	0.221	0.390	0.516	0.620	0.695	0.753	0.790	0.83	0.861	0.89	0.925	0.94	0.95	0.975	0.975
9	+	0.322	0.740	1.27	1.92	2.66	3.60	4.72	5.87	7.22	8.75	12.2	16.3	21.1	26.5	32.3
	−	0.244	0.425	0.560	0.660	0.730	0.780	0.82	0.86	0.880	0.92	0.94	0.96	0.97	0.985	0.99
10	+	0.375	0.850	1.47	2.23	3.16	4.27	5.60	7.10	8.75	10.6	14.8	19.9	25.7	32.3	39.6
	−	0.275	0.460	0.595	0.695	0.760	0.81	0.84	0.886	0.895	0.95	0.96	0.98	0.99	1.00	1.00

15	+	0.595	1.460	2.70	4.26	6.28	8.73	11.7	14.8	18.6	22.9	32.2	43.3	57.1	71.5	87.5
	−	0.374	0.595	0.735	0.810	0.86	0.898	0.93	0.945	0.95	0.98	0.98	1.00	1.00	1.00	1.00
20	+	0.850	2.22	4.31	7.02	10.6	14.8	19.9	25.7	32.2	39.8	56.5	76.5	100.0	126.0	155.0
	−	0.461	0.69	0.812	0.87	0.915	0.93	0.94	0.97	0.98	1.00	1.00	1.00	1.00	1.00	1.00
25	+	1.14	3.20	6.35	10.5	16.0	22.6	30.6	39.5	50.0	61.7	88.0	119.0	156.0	197.0	244.0
	−	0.535	0.765	0.868	0.915	0.96	0.955	0.96	0.99	1.00	1.00	1.00	1.00	1.00	1.00	1.00
30	+	1.47	4.30	8.75	14.8	22.6	32.1	44.0	56.6	71.5	88.2	126.0	171.0	225.0	283.0	350.0
	−	0.60	0.81	0.91	0.93	0.99	0.99	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
40	+	2.25	7.10	15.0	25.5	39.4	56.4	77.2	100.0	127.0	158.0	222.0	3.04	399.0	504.0	621.0
	−	0.695	0.888	0.955	0.97	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
50	+	3.21	10.6	22.9	39.7	61.0	87.5	120.0	155.0	198.0	244.0	346.0	475.0	622.0	785.0	970.0
	−	0.772	0.92	0.982	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
60	+	4.31	14.9	32.4	57.00	87.5	105.0	172.0	222.0	282.0	350.0	503.0	688.0	898.0	1130.0	1400.0
	−	0.812	0.955	0.985	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
80	+	7.12	25.8	57.0	100.0	155.0	223.0	306.0	396.0	505.0	620.0	890.0	1215.0	1585.0	2000.0	2480.0
	−	0.90	0.98	0.995	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
100	+	10.60	39.6	88.5	155.0	240.00	347.0	476.0	625.0	785.0	967.0	1395.0	1900.0	2490.0	3140.0	3870.00
	−	0.915	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00

TABLE 3 PRESSURE VARIATIONS FOR OPENING OR CLOSING TIME OF 2 SEC.

+ Sign Indicates Pressure Rise; − Sign Pressure Drop. To Get Results in Per Cent Multiply Tabular Values by 100.

$\frac{L}{H}$		Velocities in Penstock in Ft. per Sec.														
		1 ft.	2 ft.	3 ft.	4 ft.	5 ft.	6 ft.	7 ft.	8 ft.	9 ft.	10 ft.	12 ft.	14 ft.	16 ft.	18 ft.	20 ft.
1	+	0.016	0.032	0.049	0.064	0.081	0.098	0.116	0.132	0.150	0.169	0.205	0.243	0.283	0.322	0.372
	−	0.016	0.032	0.047	0.060	0.075	0.090	0.104	0.116	0.130	0.144	0.172	0.195	0.220	0.244	0.274
1.5	+	0.024	0.047	0.073	0.096	0.123	0.150	0.165	0.205	0.234	0.260	0.322	0.386	0.446	0.520	0.587
	−	0.024	0.045	0.068	0.088	0.109	0.131	0.142	0.169	0.189	0.206	0.244	0.279	0.309	0.342	0.391
2	+	0.033	0.065	0.098	0.132	0.169	0.207	0.243	0.280	0.322	0.360	0.450	0.543	0.638	0.740	0.840
	−	0.032	0.060	0.090	0.116	0.146	0.171	0.195	0.219	0.244	0.265	0.311	0.351	0.388	0.425	0.456
2.5	+	0.041	0.081	0.123	0.168	0.214	0.270	0.310	0.361	0.416	0.470	0.590	0.716	0.850	0.990	1.140
	−	0.039	0.075	0.109	0.143	0.176	0.215	0.237	0.265	0.294	0.320	0.370	0.415	0.460	0.497	0.534
3	+	0.049	0.098	0.150	0.205	0.262	0.322	0.386	0.450	0.520	0.588	0.740	0.905	1.080	1.260	1.470
	−	0.047	0.090	0.130	0.169	0.209	0.244	0.277	0.310	0.342	0.368	0.425	0.471	0.518	0.560	0.598
3.5	+	0.058	0.116	0.178	0.243	0.311	0.385	0.460	0.544	0.625	0.708	0.90	1.110	1.330	1.550	1.860
	−	0.054	0.104	0.151	0.195	0.239	0.276	0.315	0.350	0.385	0.415	0.475	0.528	0.580	0.612	0.655
4	+	0.066	0.132	0.205	0.284	0.364	0.453	0.549	0.635	0.740	0.840	1.070	1.330	1.620	1.910	2.260
	−	0.062	0.116	0.171	0.222	0.266	0.312	0.350	0.386	0.426	0.456	0.522	0.572	0.615	0.655	0.693
4.5	+	0.073	0.150	0.233	0.323	0.416	0.520	0.625	0.736	0.860	0.985	1.260	1.570	1.910	2.280	2.690
	−	0.068	0.130	0.189	0.244	0.294	0.543	0.385	0.425	0.463	0.495	0.556	0.612	0.655	0.692	0.728
5	+	0.081	0.168	0.261	0.364	0.471	0.592	0.716	0.849	0.988	1.130	1.460	1.860	2.240	2.690	3.190
	−	0.075	0.144	0.209	0.266	0.320	0.371	0.415	0.460	0.496	0.531	0.595	0.654	0.693	0.730	0.758
6	+	0.098	0.206	0.322	0.450	0.590	0.739	0.90	1.080	1.260	1.460	1.90	2.44	2.990	3.60	4.310
	−	0.090	0.171	0.244	0.310	0.360	0.425	0.475	0.515	0.558	0.590	0.625	0.712	0.758	0.782	0.812
7	+	0.116	0.243	0.385	0.540	0.715	0.902	1.110	1.320	1.58	1.82	2.40	3.06	3.82	4.660	5.60
	−	0.104	0.195	0.277	0.350	0.420	0.478	0.525	0.570	0.612	0.640	0.712	0.746	0.792	0.822	0.840
8	+	0.134	0.285	0.450	0.640	0.848	1.080	1.33	1.620	1.910	2.24	2.97	3.84	4.81	5.88	7.12
	−	0.118	0.221	0.312	0.390	0.460	0.516	0.570	0.620	0.655	0.695	0.753	0.790	0.830	0.861	0.890
9	+	0.150	0.322	0.52	0.74	0.99	1.27	1.57	1.92	2.28	2.66	3.60	4.72	5.87	7.22	8.75
	−	0.130	0.244	0.342	0.425	0.496	0.560	0.612	0.660	0.693	0.730	0.780	0.820	0.860	0.880	0.920
10	+	0.169	0.375	0.594	0.850	1.140	1.470	1.840	2.230	2.69	3.160	4.27	5.60	7.10	8.75	10.60
	−	0.144	0.275	0.372	0.460	0.533	0.595	0.650	0.695	0.730	0.760	0.810	0.840	0.886	0.895	0.950

15	+	0.264	0.595	0.99	1.46	2.02	2.70	3.130	4.260	5.25	6.280	8.73	11.70	14.80	18.60	22.90
	−	0.210	0.374	0.497	0.595	0.665	0.735	0.755	0.810	0.840	0.860	0.898	0.930	0.945	0.950	0.980
20	+	0.361	0.850	1.46	2.22	3.16	4.31	5.60	7.020	8.750	10.60	14.80	19.90	25.70	32.20	39.80
	−	0.265	0.461	0.60	0.69	0.76	0.812	0.84	0.87	0.895	0.915	0.930	0.940	0.970	0.980	1.00
25	+	0.473	1.140	2.04	3.20	4.58	6.35	8.30	10.50	13.20	16.00	22.60	30.60	39.50	50.00	61.70
	−	0.322	0.535	0.672	0.765	0.816	0.868	0.898	0.915	0.928	0.960	0.960	0.970	0.990	1.00	1.00
30	+	0.59	1.47	2.69	4.30	6.30	8.75	11.70	14.80	18.60	22.6	32.10	44.0	56.60	71.50	88.2
	−	0.371	0.60	0.73	0.81	0.853	0.910	0.950	0.955	0.960	0.99	0.99	1.00	1.00	1.00	1.00
40	+	0.852	2.20	4.32	7.10	10.70	15.00	19.90	25.50	32.2	39.40	56.40	77.2	100.0	127.0	158.0
	−	0.456	0.695	0.813	0.88	0.92	0.955	0.970	0.97	0.98	1.00	1.00	1.00	1.00	1.00	1.00
50	+	1.14	3.21	6.32	10.6	16.00	22.90	30.80	39.7	50.0	61.00	87.5	120.0	155.0	198.0	244.0
	−	0.535	0.772	0.877	0.92	0.952	0.982	0.990	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
60	+	1.46	4.31	8.50	14.9	22.60	32.40	43.50	57.00	71.5	87.5	125.0	172.0	222.0	282.0	350.0
	−	0.590	0.812	0.920	0.953	0.980	0.985	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
80	+	2.260	7.120	14.90	25.8	39.60	57.00	77.2	100.0	126.0	155.0	223.0	306.0	396.0	505.0	620.0
	−	0.695	0.90	0.95	0.98	0.995	0.995	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
100	+	3.19	10.60	22.90	39.6	61.80	88.5	120.0	155.0	197.0	240.0	347.0	476.0	625.0	785.0	967.0
	−	0.758	0.915	0.985	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00

TABLE 4 PRESSURE VARIATIONS FOR OPENING OR CLOSING TIME OF 3 SEC.

+ Sign Indicates Pressure Rise; — Sign Pressure Drop. To Get Results in Per Cent Multiply Tabular Values by 100.

$\frac{L}{H}$		Velocities in Penstock in Ft. per Sec.														
		1 ft.	2 ft.	3 ft.	4 ft.	5 ft.	6 ft.	7 ft.	8 ft.	9 ft.	10 ft.	12 ft.	14 ft.	16 ft.	18 ft.	20 ft.
1	+	0.010	0.022	0.032	0.043	0.053	0.064	0.077	0.087	0.098	0.109	0.132	0.158	0.182	0.205	0.231
	−	0.010	0.020	0.032	0.041	0.051	0.060	0.071	0.081	0.090	0.099	0.116	0.136	0.154	0.172	0.187
1.5	+	0.016	0.032	0.047	0.064	0.081	0.096	0.116	0.132	0.150	0.168	0.205	0.243	0.282	0.322	0.363
	−	0.015	0.031	0.045	0.060	0.075	0.088	0.104	0.116	0.131	0.143	0.169	0.195	0.220	0.244	0.266
2	+	0.022	0.043	0.065	0.087	0.109	0.132	0.158	0.182	0.207	0.231	0.280	0.337	0.392	0.450	0.515
	−	0.020	0.041	0.060	0.081	0.098	0.116	0.136	0.154	0.171	0.187	0.219	0.252	0.282	0.311	0.337
2.5	+	0.026	0.053	0.081	0.109	0.138	0.168	0.198	0.231	0.270	0.296	0.361	0.435	0.515	0.590	0.670
	−	0.025	0.051	0.075	0.099	0.122	0.143	0.166	0.187	0.215	0.229	0.265	0.304	0.337	0.370	0.400
3	+	0.032	0.064	0.098	0.132	0.168	0.205	0.243	0.282	0.322	0.363	0.450	0.540	0.639	0.740	0.844
	−	0.031	0.060	0.090	0.116	0.144	0.169	0.195	0.220	0.244	0.266	0.310	0.350	0.389	0.425	0.459
3.5	+	0.037	0.077	0.116	0.158	0.198	0.243	0.287	0.337	0.385	0.435	0.544	0.657	0.775	0.900	1.040
	−	0.036	0.071	0.104	0.136	0.166	0.193	0.224	0.252	0.276	0.304	0.350	0.395	0.438	0.475	0.520
4	+	0.043	0.087	0.132	0.182	0.231	0.284	0.337	0.392	0.453	0.515	0.635	0.780	0.922	1.070	1.240
	−	0.041	0.081	0.118	0.154	0.187	0.220	0.252	0.282	0.312	0.337	0.386	0.439	0.482	0.522	0.560
4.5	+	0.047	0.098	0.150	0.206	0.270	0.323	0.336	0.454	0.520	0.595	0.736	0.902	1.080	1.260	1.470
	−	0.045	0.090	0.130	0.171	0.215	0.244	0.279	0.311	0.343	0.374	0.425	0.478	0.515	0.556	0.600
5	+	0.053	0.109	0.168	0.231	0.296	0.364	0.435	0.515	0.592	0.670	0.848	1.040	1.240	1.460	1.700
	−	0.051	0.098	0.144	0.187	0.229	0.266	0.304	0.337	0.371	0.400	0.460	0.510	0.560	0.595	0.617
6	+	0.064	0.132	0.206	0.282	0.375	0.450	0.540	0.639	0.739	0.844	1.080	1.330	1.620	1.900	2.220
	−	0.060	0.116	0.171	0.220	0.275	0.310	0.350	0.389	0.425	0.459	0.515	0.572	0.620	0.652	0.690
7	+	0.077	0.159	0.243	0.337	0.435	0.540	0.657	0.775	0.902	1.040	1.320	1.670	2.02	2.40	2.86
	−	0.071	0.138	0.195	0.252	0.304	0.350	0.395	0.438	0.478	0.509	0.570	0.628	0.665	0.712	0.743
8	+	0.087	0.181	0.285	0.392	0.511	0.640	0.775	0.922	1.080	1.240	1.620	2.020	2.50	2.97	3.54
	−	0.081	0.152	0.221	0.282	0.336	0.390	0.438	0.482	0.516	0.560	0.620	0.665	0.718	0.753	0.780
9	+	0.098	0.205	0.322	0.454	0.595	0.740	0.905	1.080	1.270	1.470	1.920	2.40	2.97	3.60	4.29
	−	0.090	0.169	0.244	0.311	0.374	0.425	0.471	0.515	0.560	0.600	0.660	0.712	0.753	0.780	0.810
10	+	0.109	0.231	0.375	0.515	0.670	0.850	1.040	1.240	1.470	1.700	2.230	2.860	3.54	4.27	5 15
	−	0.099	0.187	0.275	0.337	0.400	0.460	0.510	0.560	0.595	0.617	0.695	0.735	0.780	0.810	0 942

15	+	0.168	0.363	0.595	0.844	1.140	1.460	1.860	2.200	2.700	3.210	4.260	5.60	7.10	8.73	10.60
	−	0.143	0.266	0.374	0.459	0.525	0.595	0.654	0.690	0.735	0.772	0.810	0.840	0.887	0.898	0.915
20	+	0.231	0.511	0.850	1.240	1.700	2.22	2.86	3.540	4.310	5.150	7.02	9.40	12.00	14.80	18.20
	−	0.187	0.336	0.461	0.560	0.617	0.690	0.735	0.780	0.812	0.842	0.870	0.890	0.947	0.950	0.980
25	+	0.296	0.670	0.140	1.710	2.380	3.200	4.120	5.150	6.350	7.650	10.50	14.20	18.20	22.60	28.00
	−	0.229	0.400	0.535	0.629	0.700	0.765	0.800	0.842	0.868	0.885	0.915	0.940	0.98	0.99	1.00
30	+	0.363	0.843	1.470	2.220	3.210	4.300	5.60	7.100	8.750	10.60	14.80	19.90	25.50	32.10	39.60
	−	0.266	0.462	0.600	0.690	0.772	0.81	0.840	0.887	0.910	0.915	0.930	0.98	1.00	1.00	1.00
40	+	0.515	1.240	2.250	3.540	5.120	7.100	9.40	12.00	15.00	18.20	25.50	35.30	45.00	56.40	70.00
	−	0.337	0.552	0.695	0.78	0.840	0.888	0.890	0.947	0.955	0.980	0.980	1.00	1.00	1.00	1.00
50	+	0.67	1.70	3.21	5.15	7.65	10.60	14.20	18.20	22.90	28.00	39.70	54.00	70.00	87.50	109.0
	−	0.40	0.617	0.772	0.842	0.885	0.92	0.94	0.98	0.982	0.990	1.00	1.00	1.00	1.00	1.00
60	+	0.844	2.22	4.31	7.10	10.60	14.90	19.90	25.50	32.40	39.60	57.00	77.20	100.0	25.0	54.0
	−	0.459	0.689	0.812	0.887	0.915	0.955	0.98	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
80	+	1.240	3.53	7.12	12.00	18.20	25.80	35.30	45.0	57.0	70.0	100.0	137.0	178.0	223.0	276.0
	−	0.56	0.78	0.90	0.947	0.98	0.99	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
100	+	1.70	5.120	10.60	18.20	28.00	39.6	54.00	70.00	88.50	109.0	155.0	212.0	282.0	347.0	430.0
	−	0.617	0.84	0.915	0.98	0.990	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00

TABLE 5 PRESSURE VARIATIONS FOR OPENING OR CLOSING TIME OF 4 SEC.

+ Sign Indicates Pressure Rise; — Sign Pressure Drop. To Get Results in Per Cent Multiply Tabular Values by 100.

$\frac{L}{H}$		Velocities in Penstock in Ft. per Sec.														
		1 ft.	2 ft.	3 ft.	4 ft.	5 ft.	6 ft.	7 ft.	8 ft.	9 ft.	10 ft.	12 ft.	14 ft.	16 ft.	18 ft.	20 ft.
1	+	0.008	0.016	0.024	0.032	0.041	0.049	0.058	0.064	0.073	0.081	0.098	0.116	0.132	0.150	0.169
	−	0.008	0.016	0.023	0.032	0.039	0.047	0.054	0.060	0.068	0.075	0.090	0.104	0.116	0.130	0.144
1.5	+	0.012	0.024	0.036	0.047	0.060	0.073	0.087	0.096	0.109	0.123	0.150	0.165	0.205	0.234	0.260
	−	0.012	0.024	0.035	0.045	0.056	0.068	0.081	0.088	0.099	0.109	0.131	0.142	0.169	0.189	0.206
2	+	0.016	0.033	0.047	0.065	0.081	0.098	0.016	0.132	0.150	0.169	0.207	0.243	0.280	0.322	0.360
	−	0.015	0.032	0.045	0.060	0.075	0.090	0.104	0.116	0.130	0.146	0.171	0.195	0.219	0.244	0.265
2.5	+	0.020	0.041	0.060	0.081	0.102	0.123	0.145	0.168	0.189	0.214	0.270	0.310	0.361	0.416	0 470
	−	0.019	0.039	0.056	0.075	0.093	0.109	0.126	0.143	0.159	0.176	0.215	0.237	0.265	0.294	0.320
3	+	0.024	0.049	0.073	0.098	0.123	0.150	0.178	0.205	0.233	0.262	0.322	0.386	0.450	0.520	0.588
	−	0.023	0.047	0.068	0.090	0.109	0.130	0.151	0.169	0.189	0.209	0.244	0.277	0.310	0.342	0.368
3.5	+	0.028	0.058	0.087	0.116	0.145	0.178	0.211	0.243	0.279	0.311	0.285	0.460	0.544	0.625	0.708
	−	0.027	0.054	0.081	0.104	0.126	0.151	0.175	0.193	0.216	0.239	0.276	0.315	0.350	0.385	0.415
4	+	0.032	0.066	0.098	0.132	0.169	0.205	0.243	0.284	0.322	0.364	0.453	0.544	0.635	0.740	0.840
	−	0.031	0.062	0.090	0.116	0.146	0.171	0.195	0.220	0.244	0.266	0.312	0.350	0.386	0.426	0.456
4.5	+	0.036	0.073	0.109	0.150	0.189	0.233	0.279	0.323	0.367	0.416	0.520	0.625	0.736	0.860	0.985
	−	0.035	0.068	0.099	0.130	0.159	0.189	0.216	0.244	0.266	0.294	0.343	0.385	0.425	0.463	0.495
5	+	0.041	0.081	0.123	0.168	0.214	0.261	0.310	0.364	0.414	0.471	0.592	0.716	0.848	0.988	1.130
	−	0.030	0.075	0.109	0.144	0.176	0.209	0.236	0.266	0.289	0.320	3.71	0.415	0.460	0.496	0.531
6	+	0.047	0.098	0.150	0.206	0.262	0.322	0.385	0.450	0.520	0.590	0.739	0.900	1.080	1.260	1.460
	−	0.045	0.090	0.130	0.171	0.209	0.244	0.277	0.310	0.342	0.360	0.425	0.475	0.515	0.558	0.590
7	+	0.058	0.116	0.178	0.243	0.310	0.385	0.460	0.540	0.625	0.715	0.902	1.110	1.320	1.580	1.820
	−	0.054	0.104	0.151	0.195	0.236	0.277	0.315	0.350	0.385	0.420	0.478	0.525	0.570	0.612	0.640
8	+	0.064	0.134	0.206	0.285	0.264	0.450	0.540	0.640	0.740	0.848	1.080	1.330	1.620	1.910	2.240
	−	0.060	0.118	0.171	0.221	0.266	0.312	0.350	0.390	0.426	0.460	0.516	0.570	0.620	0.655	0.695
9	+	0.073	0.150	0.233	0.322	0.414	0.520	0.625	0.740	0.860	0.990	1.270	1.570	1.920	2.28	2.66
	−	0.068	0.130	0.189	0.244	0.289	0.342	0.385	0.425	0.463	0.496	0.560	0.612	0.660	0.693	0.730
10	+	0.081	0.169	0.262	0.375	0.471	0.594	0.715	0.850	0.990	1.140	1.470	1.840	2.23	2.69	3.160
	−	0.075	0.144	0.209	0.275	0.320	0.372	0.420	0.460	0.497	0.530	0.595	0.650	0.695	0.730	0.760

15	+	0.125	0.264	0.414	0.595	0.780	0.990	1.220	1.46	1.70	2.02	2.70	3.130	4.260	5.25	6.28
	−	0.109	0.210	0.289	0.374	0.439	0.497	0.547	0.595	0.617	0.665	0.735	0.755	0.810	0.840	0.86
20	+	0.169	0.361	0.590	0.850	1.140	1.460	1.84	2.22	2.690	3.16	4.31	5.60	7.02	8.75	10.60
	−	0.146	0.265	0.360	0.461	0.533	0.600	0.650	0.690	0.730	0.760	0.812	0.84	0.87	0.895	0.915
25	+	0.214	0.473	0.780	1.140	1.550	2.040	2.560	3.20	3.82	4.58	6.35	8.30	10.50	13.20	16.00
	−	0.176	0.322	0.439	0.535	0.670	0.672	0.713	0.765	0.791	0.816	0.868	0.898	0.915	0.928	0.96
30	+	0.262	0.590	0.990	1.470	2.04	2.69	3.450	4.30	5.25	6.300	8.75	11.70	14.80	18.60	22.60
	−	0.209	0.371	0.497	0.60	0.672	0.730	0.777	0.810	0.840	0.853	0.910	0.950	0.955	0.960	0.99
40	+	0.364	0.852	1.460	2.25	3.190	4.32	3.60	7.10	8.50	10.70	15.00	19.90	25.50	32.20	39.40
	−	0.266	0.456	0.60	0.695	0.758	0.813	0.84	0.888	0.920	0.920	0.955	0.970	0.98	0.990	1.00
50	+	0.471	1.140	2.040	3.210	4.58	6.32	8.30	10.60	13.20	16.00	22.80	30.80	39.70	50.00	61.00
	−	0.320	0.535	0.672	0.772	0.815	0.877	0.898	0.920	0.928	0.95	0.982	1.99	1.00	1.00	1.00
60	+	0.590	1.460	2.690	4.31	6.32	8.50	11.70	14.90	18.60	22.60	32.40	43.50	57.00	71.50	87.50
	−	0.360	0.590	0.730	0.812	0.877	0.920	0.950	0.960	0.945	0.980	0.985	1.00	1.00	1.00	1.00
80	+	0.848	2.260	4.320	7.120	0.60	19.90	19.90	25.8	32.20	39.60	57.0	77.20	100.0	126.0	155.0
	−	0.460	0.695	0.813	0.90	0.915	0.955	0.970	0.98	0.99	0.995	0.995	1.00	1.00	1.00	1.00
100	+	1.140	3.190	6.320	10.60	16.10	22.90	30.80	39.60	50.00	81.80	88.5	120.0	155.0	197.0	240.0
	−	0.533	0.758	0.877	0.915	0.950	0.985	0.990	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00

TABLE 6 PRESSURE VARIATIONS FOR OPENING OR CLOSING TIME OF 5 SEC.

+ Sign Indicates Pressure Rise; − Sign Pressure Drop. To Get Results in Per Cent Multiply Tabular Values by 100.

$\frac{L}{H}$		Velocities in Penstock in Ft. per Sec.														
		1 ft.	2 ft.	3 ft.	4 ft.	5 ft.	6 ft.	7 ft.	8 ft.	9 ft.	10 ft.	12 ft.	14 ft.	16 ft.	18 ft.	20 ft.
1	+	0.006	0.013	0.019	0.024	0.032	0.038	0.045	0.051	0.058	0.064	0.078	0.091	0.104	0.118	0.132
	−	0.006	0.013	0.019	0.024	0.032	0.037	0.043	0.049	0.054	0.060	0.072	0.083	0.095	0.106	0.116
1.5	+	0.009	0.019	0.028	0.038	0.047	0.058	0.068	0.078	0.087	0.096	0.118	0.139	0.161	0.183	0.205
	−	0.009	0.019	0.028	0.037	0.045	0.054	0.064	0.072	0.081	0.088	0.106	0.112	0.138	0.154	0.169
2	+	0.013	0.024	0.038	0.051	0.065	0.078	0.091	0.104	0.118	0.132	0.161	0.189	0.221	0.250	0.280
	−	0.013	0.024	0.037	0.049	0.060	0.072	0.083	0.095	0.106	0.116	0.138	0.159	0.181	0.200	0.219
2.5	+	0.016	0.032	0.047	0.064	0.081	0.095	0.116	0.132	0.150	0.168	0.205	0.243	0.285	0.322	0.361
	−	0.016	0.032	0.045	0.060	0.075	0.090	0.104	0.116	0.130	0.143	0.169	0.195	0.221	0.244	0.265
3	+	0.019	0.038	0.058	0.078	0.098	0.118	0.139	0.161	0.183	0.205	0.250	0.299	0.346	0.399	0.450
	−	0.019	0.037	0.054	0.072	0.090	0.106	0.112	0.138	0.154	0.169	0.200	0.231	0.256	0.284	0.310
3.5	+	0.022	0.045	0.068	0.091	0.116	0.139	0.165	0.189	0.214	0.243	0.299	0.266	0.414	0.475	0.544
	−	0.022	0.043	0.064	0.083	0.104	0.112	0.141	0.159	0.176	0.193	0.231	0.262	0.289	0.323	0.350
4	+	0.024	0.051	0.078	0.104	0.132	0.161	0.189	0.221	0.250	0.284	0.346	0.414	0.485	0.561	0.635
	−	0.024	0.049	0.072	0.092	0.116	0.138	0.159	0.181	0.200	0.220	0.256	0.289	0.326	0.359	0.386
4.5	+	0.028	0.058	0.087	0.118	0.150	0.183	0.214	0.250	0.285	0.323	0.399	0.475	0.561	0.650	0.736
	−	0.028	0.054	0.081	0.106	0.130	0.154	0.175	0.200	0.222	0.244	0.284	0.327	0.259	0.396	0.425
5	+	0.032	0.64	0.098	0.132	0.168	0.205	0.243	0.285	0.322	0.364	0.446	0.540	0.640	0.740	0.848
	−	0.032	0.060	0.090	0.116	0.144	0.169	0.195	0.221	0.244	0.266	0.309	0.350	0.390	0.425	0.460
6	+	0.038	0.078	0.118	0.161	0.206	0.250	0.299	0.346	0.399	0.450	0.561	0.678	0.800	0.932	1.080
	−	0.037	0.072	0.106	0.138	0.171	0.200	0.231	0.256	0.284	0.310	0.359	0.405	0.445	0.483	0.515
7	+	0.045	0.091	0.139	0.189	0.243	0.299	0.366	0.414	0.475	0.540	0.678	0.822	0.988	1.150	1.320
	−	0.043	0.083	0.112	0.159	0.195	0.231	0.262	0.289	0.321	0.350	0.405	0.451	0.496	0.537	0.570
8	+	0.051	0.104	0.161	0.221	0.285	0.346	0.414	0.485	0.561	0.640	0.800	0.988	1.180	1.390	1.620
	−	0.049	0.095	0.138	0.181	0.221	0.256	0.289	0.326	0.359	0.390	0.445	0.496	0.542	0.580	0.620
9	+	0.058	0.118	0.183	0.250	0.322	0.399	0.475	0.561	0.650	0.740	0.932	1.150	1.390	1.630	1.920
	−	0.054	0.106	0.154	0.200	0.244	0.284	0.321	0.359	0.396	0.425	0.483	0.537	0.580	0.615	0.660
10	+	0.064	0.132	0.206	0.283	0.375	0.446	0.540	0.640	0.740	0.850	1.080	1.330	1.620	1.920	2.230
	−	0.060	0.116	0.171	0.220	0.275	0.309	0.350	0.390	0.425	0.460	0.516	0.572	0.618	0.660	0.695

15	+	0.098	0.206	0.322	0.446	0.595	0.740	0.902	1.080	1.270	1.460	1.920	2.410	2.990	3.530	4.260
	−	0.090	0.171	0.244	0.309	0.374	0.425	0.478	0.520	0.560	0.595	0.660	0.707	0.748	0.780	0.810
20	+	0.132	0.285	0.450	0.638	0.850	1.080	1.320	1.620	1.920	2.220	2.990	3.82	4.800	5.87	7.020
	−	0.116	0.221	0.310	0.388	0.461	0.516	0.570	0.618	0.660	0.690	0.748	0.791	0.820	0.860	0.870
25	+	0.168	0.375	0.595	0.850	1.140	1.470	1.820	2.250	2.660	3.200	4.310	5.600	7.120	8.750	10.50
	−	0.143	0.275	0.374	0.460	0.535	0.600	0.640	0.691	0.730	0.765	0.812	0.840	0.885	0.898	0.915
30	+	0.206	0.450	0.740	1.080	1.470	1.920	2.410	2.990	3.530	4.30	5.870	7.750	9.900	12.20	14.80
	−	0.171	0.310	0.425	0.518	0.600	0.660	0.707	0.748	0.780	0.810	0.860	0.892	0.917	0.940	0.950
40	+	0.285	0.640	1.080	1.620	2.250	2.990	3.820	4.800	5.870	7.100	9.900	13.10	16 80	21.00	25.50
	−	0.221	0.390	0.518	0.615	0.695	0.758	0.791	0.820	0.860	0.888	0.917	0.940	0 950	0.970	0.98
50	+	0.375	0.850	1.470	2.240	3.210	4.310	5.600	7.120	8.750	10.60	14.90	19.90	25 70	32.40	39.70
	−	0.275	0.461	0.600	0.693	0.772	0.812	0.840	0.885	0.898	0.920	0.950	0.98	0 965	0.980	1.00
60	+	0.540	1.320	1.920	2.990	4.310	5.870	7.750	9.900	12.20	14.90	21.00	28.40	36.60	46.20	57.00
	−	0.350	0.570	0.660	0.758	0.812	0.860	0.892	0.917	0.940	0.955	0.970	0.995	1.00	1.00	1.00
80	+	0.640	1.620	2.990	4.810	7.120	9.80	13.10	16.80	21.00	25.80	36.60	49.90	64.50	81.00	100.0
	−	0.390	0.620	0.758	0.83	0.900	0.925	0.940	0.940	0.970	0.98	0.985	1.00	1.00	1.00	1.00
100.	+	0.850	2.25	4.310	7.100	10.60	14.90	19.90	25.70	32.4	39.50	56.70	76.5	100.0	126.0	155.0
	−	0.461	0.695	0.812	0.885	0.915	0.955	0.98	0.965	0.980	1.00	1.00	1.00	1.00	1.00	1.00

TABLE 7 PRESSURE VARIATIONS FOR OPENING AND CLOSING TIME OF 6 SEC.

+ Sign Indicates Pressure Rise; − Sign Pressure Drop. To Get Results in Per Cent Multiply Tabular Values by 100.

$\frac{L}{H}$		Velocities in Penstock in Ft. per Sec.														
		1 ft.	2 ft.	3 ft.	4 ft.	5 ft.	6 ft.	7 ft.	8 ft.	9 ft.	10 ft.	12 ft.	14 ft.	16 ft.	18 ft.	20 ft.
1	+	0.005	0.010	0.016	0.022	0.026	0.032	0.037	0.043	0.049	0.053	0.064	0.077	0.087	0.098	0.109
	−	0.005	0.010	0.016	0.020	0.025	0.032	0.036	0.041	0.047	0.051	0.060	0.071	0.081	0.090	0.99
1.5	+	0.008	0.016	0.024	0.032	0.041	0.047	0.058	0.064	0.073	0.081	0.096	0.116	0.132	0.150	0.168
	−	0.008	0.015	0.024	0.031	0.039	0.045	0.054	0.060	0.068	0.075	0.088	0.104	0.116	0.131	0.143
2	+	0.010	0.022	0.033	0.043	0.053	0.065	0.077	0.087	0.098	0.109	0.132	0.158	0.182	0.207	0.231
	−	0.010	0.020	0.032	0.041	0.051	0.060	0.071	0.081	0.090	0.098	0.116	0.136	0.154	0.171	0.187
2.5	+	0.013	0.026	0.041	0.053	0.068	0.081	0.098	0.109	0.123	0.138	0.168	0.198	0.231	0.270	0.296
	−	0.013	0.025	0.039	0.051	0.064	0.075	0.090	0.099	0.109	0.122	0.143	0.166	0.187	0.215	0.229
3	+	0.016	0.032	0.049	0.064	0.081	0.098	0.116	0.132	0.150	0.168	0.205	0.243	0.282	0.322	0.363
	−	0.016	0.031	0.047	0.060	0.075	0.090	0.104	0.116	0.130	0.144	0.169	0.195	0.220	0.244	0.266
3.5	+	0.018	0.037	0.058	0.077	0.098	0.116	0.139	0.158	0.178	0.198	0.243	0.287	0.337	0.385	0.435
	−	0.018	0.036	0.054	0.071	0.090	0.104	0.122	0.136	0.151	0.166	0.193	0.224	0.252	0.276	0.304
4	+	0.022	0.043	0.066	0.087	0.109	0.132	0.158	0.182	0.205	0.231	0.284	0.337	0.392	0.453	0.515
	−	0.020	0.041	0.062	0.081	0.099	0.116	0.136	0.154	0.171	0.187	0.220	0.252	0.282	0.312	0.337
4.5	+	0.024	0.047	0.073	0.098	0.123	0.150	0.181	0.206	0.233	0.270	0.323	0.386	0.454	0.520	0.595
	−	0.023	0.045	0.068	0.090	0.109	0.130	0.152	0.171	0.189	0.215	0.244	0.279	0.311	0.343	0.374
5	+	0.026	0.053	0.081	0.109	0.138	0.168	0.198	0.231	0.261	0.296	0.364	0.435	0.515	0.592	0.670
	−	0.025	0.051	0.075	0.098	0.122	0.144	0.166	0.187	0.209	0.229	0.266	0.304	0.337	0.371	0.400
6	+	0.032	0.064	0.098	0.132	0.168	0.206	0.243	0.282	0.322	0.375	0.450	0.540	0.639	0.739	0.844
	−	0.031	0.060	0.090	0.116	0.143	0.171	0.195	0.220	0.244	0.275	0.310	0.350	0.389	0.425	0.459
7	+	0.037	0.077	0.116	0.159	0.198	0.243	0.287	0.337	0.385	0.435	0.540	0.657	0.775	0.902	1.040
	−	0.036	0.071	0.104	0.138	0.166	0.195	0.224	0.252	0.277	0.304	0.350	0.395	0.438	0.478	0.509
8	+	0.043	0.087	0.134	0.181	0.231	0.285	0.337	0.392	0.450	0.511	0.640	0.775	0.922	1.080	1.240
	−	0.041	0.081	0.118	0.152	0.187	0.221	0.252	0.282	0.312	0.336	0.390	0.438	0.482	0.516	0.560
9	+	0.047	0.098	0.150	0.205	0.270	0.322	0.386	0.454	0.520	0.595	0.740	0.905	1.080	1.270	1.470
	−	0.045	0.090	0.130	0.169	0.215	0.244	0.279	0.311	0.342	0.374	0.425	0.471	0.515	0.560	0.600
10	+	0.053	0.109	0.169	0.231	0.296	0.375	0.435	0.515	0.594	0.670	0.850	0.104	1.240	1.470	1.700
	−	0.051	0.099	0.144	0.187	0.229	0.275	0.304	0.337	0.372	0.400	0.460	0.510	0.560	0.595	0.617

15	+	0.081	0.168	0.264	0.363	0.473	0.595	0.716	0.844	0.990	1.140	1.460	1.860	2.220	2.700	3.210
	−	0.075	0.143	0.210	0.266	0.322	0.374	0.415	0.459	0.497	0.525	0.595	0.654	0.690	0.735	0.772
20	+	0.109	0.231	0.361	0.511	0.672	0.850	1.040	1.240	1.460	1.700	2.220	2.86	3.540	4.310	5.150
	−	0.099	0.187	0.265	0.336	0.400	0.461	0.510	0.560	0.600	0.617	0.690	0.735	0.780	0.812	0.842
25	+	0.138	0.296	0.473	0.670	0.902	1.140	1.470	1.710	2.040	2.320	3.200	4.120	5.150	6.350	7.650
	−	0.122	0.229	0.322	0.400	0.478	0.535	0.600	0.629	0.672	0.700	0.765	0.80	0.842	0.868	0.885
30	+	0.168	0.363	0.590	0.843	1.140	1.470	1.860	2.220	2.690	3.210	4.300	5.600	7.100	8.78	10.60
	−	0.143	0.266	0.371	0.462	0.525	0.600	0.654	0.690	0.730	0.771	0.810	0.84	0.887	0.910	0.915
40	+	0.231	0.515	0.852	1.240	1.700	2.250	2.860	3.540	4.320	5.120	7.100	9.40	12.00	15.00	18.20
	−	0.187	0.337	0.456	0.552	0.617	0.695	0.735	0.780	0.813	0.840	0.888	0.89	0.947	0.955	0.980
50	+	0.296	0.670	1.140	1.700	2.380	3.210	4.120	5.150	6.320	7.650	10.60	14.20	18.20	22.90	28.00
	−	0.229	0.400	0.535	0.617	0.700	0.772	0.800	0.842	0.877	0.885	0.92	0.94	0.98	0.982	0.99
60	+	0.363	0.844	1.460	2.220	3.210	4.310	5.600	7.100	8.500	10.60	14.90	19.90	25.50	32.40	39.60
	−	0.266	0.459	0.590	0.689	0.772	0.812	0.840	0.887	0.920	0.915	0.955	0.98	1.00	1.00	1.00
80	+	0.515	1.240	2.260	3.530	5.120	7.120	7.220	12.00	14.90	18.20	25.8	35.3	45.00	57.00	70.00
	−	0.337	0.560	0.695	0.780	0.840	0.900	0.880	0.947	0.955	0.980	0.98	1.00	1.00	1.00	1.00
100	+	0.670	1.700	3.190	5.120	7.650	10.60	14.20	18.20	22.90	28.00	39.60	54.0	70.00	88.5	109.0
	−	0.400	0.617	0.758	0.840	0.885	0.915	0.940	0.980	0.985	0.990	1.00	1.00	1.00	1.00	1.00

TABLE 8 VELOCITY OF VIBRATIONS IN PENSTOCK

$$a = \sqrt{\frac{\frac{g}{\gamma}}{\frac{1}{\epsilon} + \frac{1}{E} \cdot \frac{D}{d}}} = \frac{22{,}720}{\sqrt{23.5 + K \cdot \frac{D}{d}}}$$

Material of Penstock	Dia. of Penstock	Thickness of Penstock Walls in in.									
		¼	½	¾	1	1¼	1½	2	3	4	5
Steel Plate	1 ft.	3810	4160	4330	4400	4460	4530				
Cast iron		3380	3900	4100	4240	4300	4350	4430	4500		
Wood stave			2050	2400	2660	2860	3000	3250	3570	3800	
Steel plate	2 ft.	3310	3830	4040	4170	4270	4340				
Cast iron		2800	3370	3700	3900	4000	4100	4230	4380		
Wood stave				1810	2050	2230	2400	2650	3000	3250	
Steel plate	3 ft.	2950	3570	3820	4000	4100	4150				
Cast iron		2440	3050	3370	3600	3750	3900	4030	4220		
Wood stave				1520	1720	1880	2050	2300	2650	2900	
Steel plate	4 ft.	2700	3310	3600	3810	3960	4040				
Cast iron		2190	2800	3150	3390	3580	3700	3900	4100		
Wood stave				1330	1525	1680	1700	2050	2400	2650	
Steel plate	5 ft.	2530	3120	3450	3650	3830	3900				
Cast iron		2000	2600	2970	3210	3380	3520	3750	4000		
Wood stave					1380	1530	1660	1870	2200	2450	2650
Steel plate	6 ft.	2330	2950	3320	3600	3700	3850				
Cast iron			2450	2800	3050	3250	3400	3610	3900		
Wood stave					1260	1400	1520	1720	2050	2300	2510
Steel plate	7 ft.	2200	2810	3190	3410	3590	3710				
Cast iron				2650	2920	3120	3260	3500	3800		
Wood stave					1160	1300	1500	1600	1920	2160	2360
Steel plate	8 ft.	2100	2700	3060	3300	3470	3600				
Cast iron					2790	3000	3150	3380	3700		
Wood stave						1230	1330	1520	1800	2050	2230
Steel plate	9 ft.	2000	2600	2950	3200	3400	3570				
Cast iron					2650	2900	3050	3310	3600		
Wood stave						1170	1260	1450	1730	1960	2130
Steel plate	10 ft.	1920	2540	2880	3130	3310	3450				
Cast iron						2800	2980	3220	3520		
Wood stave						1120	1200	1380	1660	1880	2050
Steel plate	12 ft.	1770	2330	2700	3000	3150	3300				
Cast iron						2620	2790	3050	3380		
Wood stave						1000	1120	1260	1520	1720	1880
Steel plate	14 ft.	1660	2200	2560	2820	3000	3190				
Cast iron						2500	2650	2920	3260		
Wood stave						935	1040	1170	1490	1610	1770
Steel plate	16 ft.	1560	2100	2450	2710	2890	3060				
Cast iron						2400	2550	2800	3150		
Wood stave						875	965	1120	1330	1520	1670
Steel plate	18 ft.	1480	2000	2350	2600	2800	2950				
Cast iron							2450	2700	3050	3300	
Wood stave							900	1050	1260	1440	1590
Steel plate	20 ft.	1410	1920	2250	2520	2700	2870				
Cast iron							2330	2600	2960	3200	
Wood stave							870	990	1200	1380	1520

DISCUSSION

H. E. Warren.[1] Until recently the company with which I am connected was obliged to depend entirely upon mathematical analysis for the solution of governing problems. When formula [3], given by Mr. Uhl, was first published about four or five years ago, it was thought that the formula would be of great

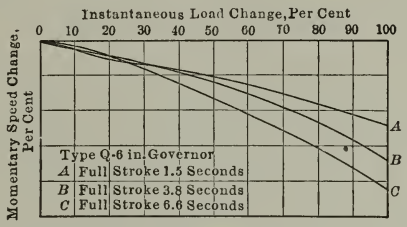


Fig. 12 Curves Plotted from Experiments, Showing Load Increase

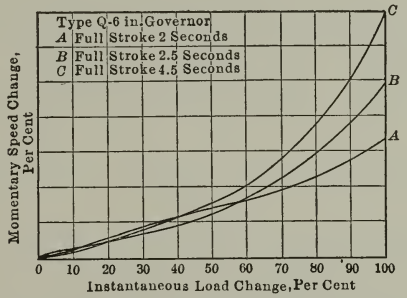


Fig. 13 Curves Plotted from Experiments, Showing Load Decrease

use. I must caution engineers against placing too much reliance on it, for, although it seems to be correct, there are too many uncertain features, such as modifications of load by momentary changes and characteristics in rate and time of action of the governor that cannot be accurately determined. In fact, the only feature which is perfectly clear is Wr^2, and even this is apt to be misleading. The flywheel effect of the generator is usually considered, but that of the motors overlooked.

The Lombard Governor Company maintains a hydraulic laboratory in its own factory, where wheels are tested under conditions which are the same as actual service. Experience has shown that with Lombard governors the regulation is much

[1] Lombard Governor Co., Ashland, Mass.

better for load changes of less than 75% than the formula would indicate. The formula is of very little use below 50%, in which case the speed regulation can be predicted only by referring to the experimental data and departing from the formula.

The curves in Figs. 12 and 13 were plotted from test data, and are quite different from those shown in Fig. 3. The curvature is in the opposite direction, that is, concave instead of convex. This shows that the speed regulation below 100% load change is very much better than the formula would indicate.

In connection with long pipe lines, it is possible to reduce the fluctuations in pressure and improve the speed regulation by attaching to the governor a device for dampening out the waves in the pipe lines which might otherwise become troublesome.

In reply to a question, Mr. Warren stated that the wheels were tested by a generator load which was as nearly as possible a duplicate of a commercial load. A generator capable of taking the full output of the wheel is used, together with very efficient switching arrangements and accurate recording instruments for the speed variation and the gate movement.

Mr. Warren also referred to the trouble which arises where wheels are not deeply submerged. In such cases whirlpools sometimes form, and permit air to affect the vacuum in the draft tube. When this occurs, regulation is out of the question.

F. M. Gunby[1] referring to trouble caused by air in the draft tubes, said that the depth of the wheelpit must be sufficient to admit and submerge the draft tube properly. He mentioned cases where the tubes sometimes became unsealed, thereby losing the draft in them. As this condition causes the loss of part, or all the fall below the center of the wheel, its effect on the regulation is very injurious. Where that part of the head below the center of the wheel is a large proportion of the total, regulation becomes impossible if the draft tubes are not properly sealed.

E. A. Ekern[2] stated that a great number of Western plants are having trouble with air in the draft tube, which varies between 1 per cent and 1½ per cent, and with the absolute pressure.

The author did not desire to present a closure.—Editor.

[1] With Chas. T. Main, Boston, Mass.

[2] Stone & Webster Engrg. Corp., Boston, Mass.

No. 1355

THE PRESENT STATE OF DEVELOPMENT OF LARGE STEAM TURBINES

By A. G. Christie, Madison, Wis.

Associate Member of the Society

The Transactions of the Society for 1910 include two papers on steam turbines, one by Sam L. Naphtaly[1] and the other by F. H. Varney,[2] which, with the accompanying discussions, give interesting information on the present development of two prominent American types. Mr. Geo. A. Orrok discussed small steam turbines in his paper published in Vol. 31 of the Transactions.[3] Vol. 25 contains papers and discussions[4] presented at the Chicago meeting of the Society in 1904, which cover the leading types of steam turbines then in commercial use. It seems fitting now to record the present state of development of large steam turbine construction and of the field of the steam turbine both in this country and in Europe.

PRESENT FIELD OF LARGE STEAM TURBINES

2 Steam turbines are now being used for driving alternating-current generators, turbo-compressors, turbo-blowers, pumps and marine propellers, and, by means of gearing, to furnish power to direct-current generators, rolling mills and the propeller shafts of steamships. Reciprocating engines were formerly used for such purposes, but recently this class of engine has seldom been installed except for rolling-mill work, non-condensing service as on heating systems, rope and belt drives, hoists, and in certain combinations with low-pressure turbines in marine work. The high economy of the piston-pumping engines and also of some types of air compres-

[1] Vol. 32, p. 1251.

[2] Vol. 32, p. 1267.

[3] Vol. 31, 1909, p. 263.

[4] A. Rateau, Different Applications of Steam Turbines, p. 782; W. L. R. Emmet, The Steam Turbine in Modern Engineering, p. 1041; E. S. Lea, The DeLaval Steam Turbine, p. 1056.

Presented at the Spring Meeting, Cleveland 1912, of The American Society of Mechanical Engineers.

sors, has continued their popularity in spite of the increasing competition of steam turbine units. The steam turbine has found favor principally on account of its low first cost of installation, its small floor space requirements, its continued good steam economy over a period of years and its small operating and repair charges.

3 The increased use of steam turbines in sizes up to 1000 h.p. seems to have received at least a temporary check in Europe by the introduction of the new Stumpf direct-flow engine.

TYPES OF STEAM TURBINES

4 For the purpose of this paper, large commercial steam turbines will be divided into two classes: (*a*) fundamental types, and (*b*) modified or combined types.

5 The fundamental types of turbines are as follows:

a The Parsons type, which works on the so-called "reaction" principle. In this type the heat energy of the steam is changed into kinetic energy, both in the stationary guide blades and in the moving blades. In other words, both sets of blades act as orifices expanding the steam through a small pressure drop. As nozzles and orifices usually have very high efficiencies, this turbine should, theoretically, prove the most economical of all types. The construction of the Parsons turbine is familiar to all engineers. It consists of a drum, or a number of drums, carrying the blade rows which alternate with rows in the casing. The drums carry balance pistons to equalize the end thrust.

b The Curtis type, which works on the impulse principle with high steam velocities and few stages. Each stage, however, is provided with two or more rows of revolving blades known as "velocity rows," with intermediate rows of guide blades. The steam velocity at the beginning of each stage is high. The characteristics of this type of turbine are presented in Mr. W. L. R. Emmet's paper, The Steam Turbine in Modern Engineering.[1] The revolving blades are carried on disks separated by diaphragms, which extend to the shaft and which carry the orifices between stages. Curtis turbines are now usually built with horizontal shafts. In American practice some sizes over 7000 kw. still have vertical shafts.

c The Rateau turbine, which consists of a number of simple impulse wheels in series on the same shaft and separated

[1] Trans. Am. Soc. M. E., vol. 25, p. 1041.

from one another by diaphragms carrying nozzles. It operates with lower steam velocities than the Curtis and consequently has many more stages. Each revolving element carries only one row of blades. The characteristics of this type will be found in Prof. A. Rateau's paper, Different Applications of Steam Turbines.[1]

The type of turbine known as the Zoelly belongs to the same classification as the Rateau, from which it differs only in the use of higher steam velocities, in the number of stages and in certain constructional details. Fig. 1 shows a section of a Zoelly turbine. Though this type has been widely adopted abroad, it has not been placed on the American market up to the present time.

6 Each of these fundamental types is based on sound theoretical principles. In the process of manufacture and in operation, certain features have not proved entirely satisfactory, hence far-reaching modifications have been made in the design of some types of turbines. Some manufacturers have combined the characteristics of two or more types to overcome the inherent limitations of each fundamental type. A discussion of the unsatisfactory operating conditions of each type will show the reasons which led up to the changes in recent turbines and will also aid in distinguishing the novel features of new designs.

7 The first rows of spindle blades in a standard Parsons turbine are placed on a drum of small diameter in order to make the blades as long as possible and to minimize the proportional leakage losses past the ends of the blades. A large number of rows are provided in order to keep the steam velocities low, as the blade velocities must be low with the small drum. The drop in pressure at each row is small and hence the leakage is correspondingly reduced. This construction results in a turbine with a long spindle and with great distance between bearings. High-pressure steam, frequently at high temperature, is admitted to the casing. Distortion of casing and spindle are thus easily conceivable in such construction, and to allow for this contingency the clearance on the ends of all blades is usually increased. This distortion may be due either to unequal heating or to the growth of the cast-iron casings. The fluid friction losses are large in this high-pressure section, for a large number of rows of blades must be revolved in steam of high density. The leakage losses and fluid friction losses in the high-pressure section, to-

[1] Trans. Am. Soc. M. E., vol. 25, p. 782.

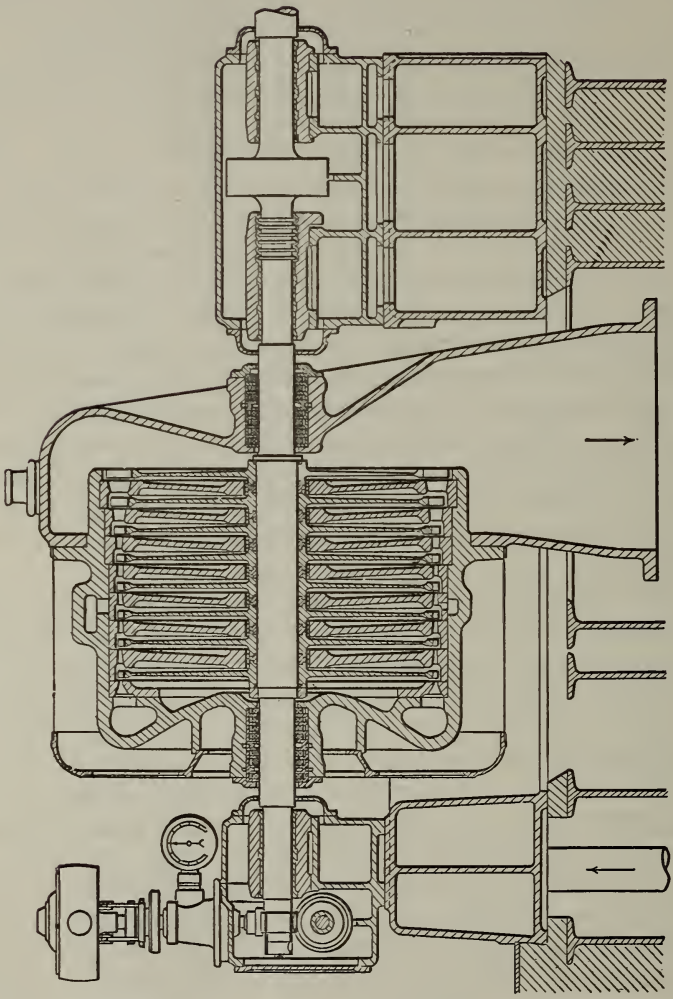


FIG. 1 SECTION OF ZOELLY TURBINE

gether with the troubles due to distortion in the long shaft, have forced designers to introduce modifications in this portion of the turbine, either by new blading arrangements, by dividing the total expansion in large sizes between two cylinders or by the introduction of impulse blading.

8 The low-pressure sections of Parsons turbines have always shown high efficiencies. As low steam velocities are characteristic of this type, there is no cutting away of blade material, even with very wet steam, provided no injurious properties are present in the feedwater. This low-pressure section has therefore been altered only in details.

9 The presence of a large low-pressure balance piston in close proximity to the steam inlet has frequently been the cause of serious distortions. Many builders now place this balance piston in the exhaust end, a construction due to Fullager.

10 The Curtis turbine utilizes high steam velocities in all stages. As steam becomes wet through expansion in the low-pressure stages, there is frequently considerable cutting of the blade material by the steam, although, contrary to first expectations, there is seldom cutting in the high-pressure blades due to the high initial velocities. The first row of velocity blades in a stage usually does the greater portion of the work, and hence the second row does not work at maximum efficiency. As the steam of the first stage is expanded very fully in the nozzles, there is no high pressure or superheat in the turbine casing or at the glands. The vertical type of unit is sometimes subject to electrical unbalancing and to other troubles peculiar to this construction. It is not as accessible in operation as the horizontal machines.

11 Recent designs have provided for horizontal units and for the replacement of the low-pressure section by sections of other types.

12 The Rateau turbine has high pressures on the gland at one end. There are a large number of disks revolving in dense steam at the high-pressure end. It has lower steam velocities throughout than the Curtis, and consequently has no blade-cutting effects in the low-pressure section. The clearances around the blades are large, but the shaft clearances of the diaphragms must be kept small. Some builders of this and the Curtis type have brought out new designs which employ the high-pressure section of the Curtis with the low-pressure section of the Rateau. These represent a compromise between efficiency and manufacturing costs.

13 Under the classification of modified or combined types, there are turbines with modified Parsons, Curtis and Rateau construction,

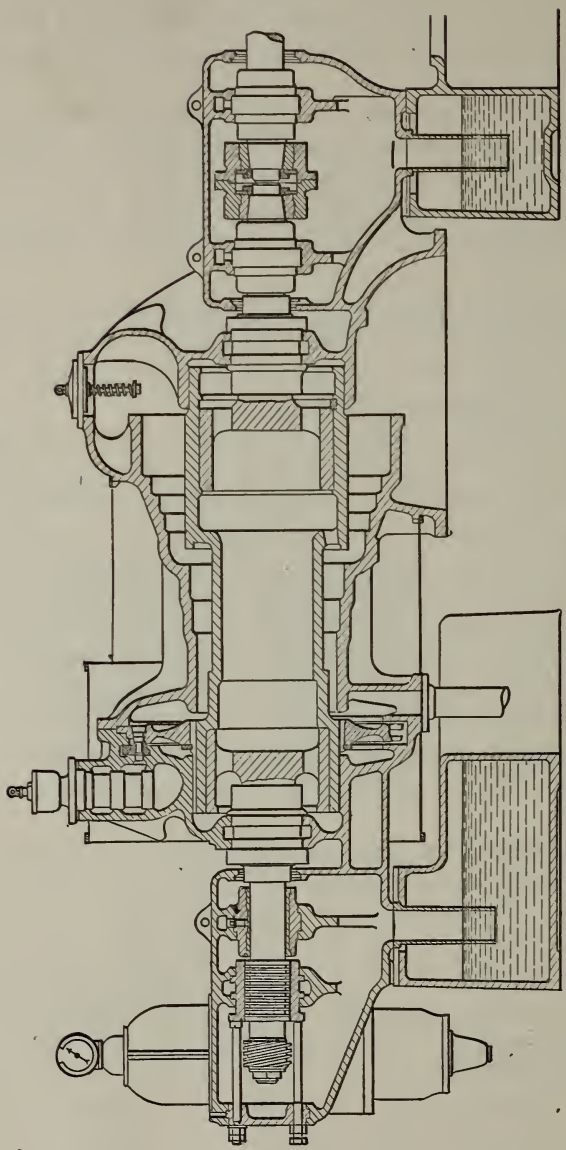


FIG. 2 SECTION OF BROWN BOVERI'S CURTIS-PARSONS TURBINE

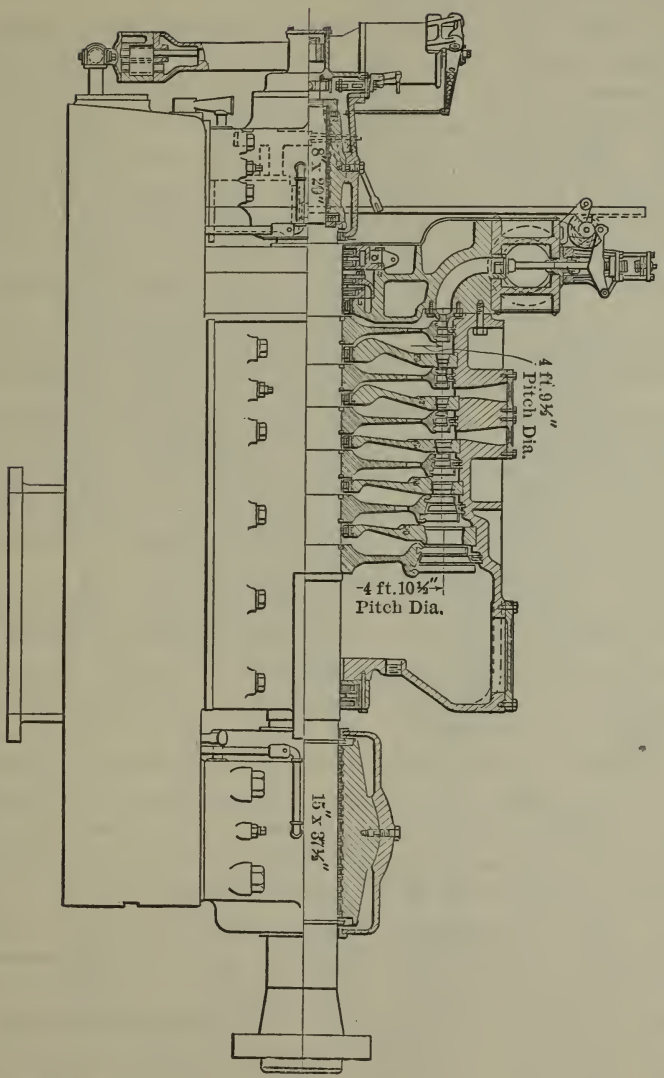


FIG. 3 HORIZONTAL CURTIS STEAM TURBINE, 7000 KW., 1800 R.P.M., SIX-STAGE

and turbines with combinations of Curtis-Parsons, Rateau-Parsons, Curtis-Rateau and Curtis-Zoelly construction, with a few special combinations. Some typical turbines of these classes are described in the following paragraphs.

14 The Allis-Chalmers Company manufacture a turbine of the modified Parsons type. High peripheral speeds are employed with a decreased number of rows. A portion of the theoretical efficiency in the high-pressure end has been sacrificed by the use of fewer blade rows. Also a smaller proportion of work is done in this section than is usual in Parsons turbines. The blades are all provided with a channel-shaped shroud. European experimenters have pointed out that better efficiencies are obtained with shrouded than with plain blades, as the so-called spilling-over at the ends is prevented. Wing blades are fitted in the last low-pressure rows to take care of the large volumes of low-pressure steam. The spindle is much shorter and stiffer than the standard Parsons, and hence smaller clearances can be provided. The Fullager low-pressure balance piston is also used. The outstanding features of this design are, reaction principle with drum construction, few rows of blades with high steam velocities, short spindle construction, and employment of wing blades for high vacuum. The steam consumptions obtained on this type show improved efficiency over standard construction.

15 A number of Parsons turbines have been built in which the total expansion is divided between two cylinders. One of the best known units of this type is installed at Dunstan Power Station, England. This was built by Richardsons Westgarth & Company on Brown-Boveri designs. This unit, however, employs a modified double-flow Parsons construction in the low-pressure cylinder.

16 C. A. Parsons & Company have used cast-steel cylinders for the high-pressure portion of their turbines so as to overcome the troubles due to deformation and growth of metal under high pressures and superheat experienced with cast-iron casings.

17 The Westinghouse Machine Company have developed a double-flow machine which employs a Curtis high-pressure stage with Parsons intermediate and double-flow low-pressure sections. There is only one balance piston in this machine next to the Curtis ring. Mr. Sam L. Naphtaly describes this type of machine in his paper, Test of a 10,000-Kw. Steam Turbine, and also shows a cross-section of the turbine.[1]

18 These turbines run at high speeds, have short shafts and small clearances. The steam is expanded in the nozzles, and hence

[1] Trans. Am. Soc. M. E., vol. 32, p. 1251.

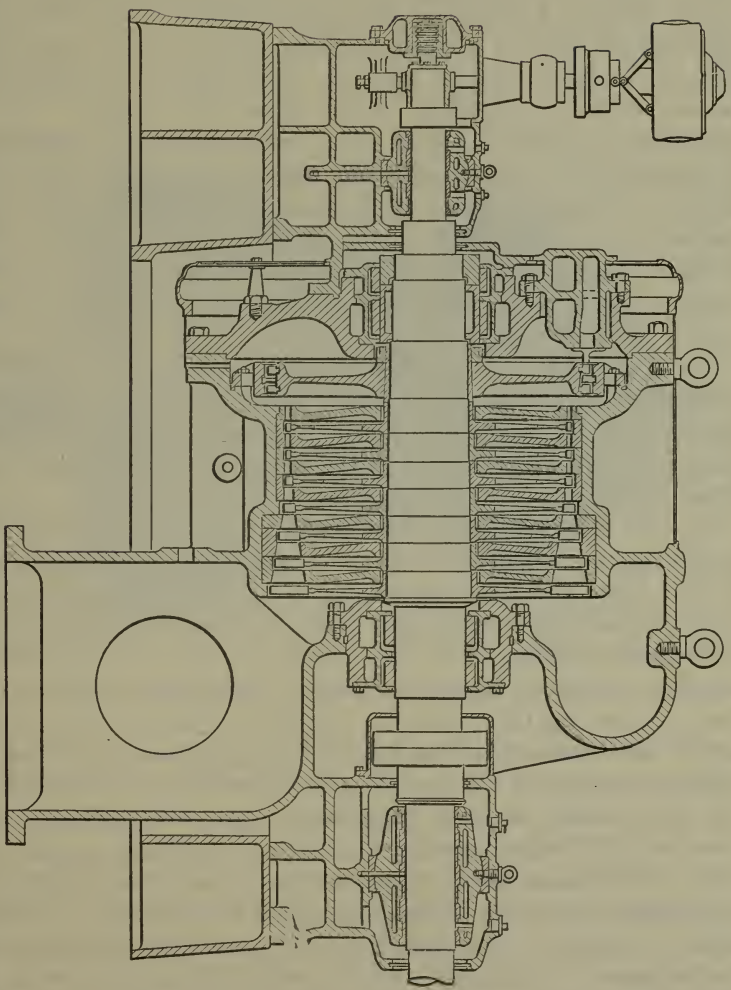


FIG. 4 SECTION OF BERGMANN CURTIS-RATEAU TURBINE

there are no high pressures or temperatures in the casing. By providing two low-pressure sections, high vacuum can be economically utilized.

19 European builders of Parsons turbines, among whom are Brown Boveri & Company, C. A. Parsons & Co., Franco Tosi, Sulzers, Willans & Robinson and Erste Brünner, have replaced the high-pressure sections of their Parsons turbines by a Curtis wheel with two or more velocity rows, but have retained the single-flow Parsons drum construction for the remaining portions. One of these units is shown in Fig. 2. The temperatures and pressures in the casing are low as the steam is expanded in the nozzles. The distance between bearings is decreased, the shaft is stiffer and clearances are smaller than in the standard Parsons turbine. The Fullager balance piston is used in the turbines of several of these builders. Turbines built in this manner have shown some exceptionally good efficiencies.

20 Compared with the Westinghouse machine, the leakage at the end of the low-pressure blading is less than in the double-flow section. On the other hand, the Westinghouse machine has smaller balance piston losses and can also utilize the highest vacuum at better efficiency.

21 The Melms & Pfenninger Company formerly employed a drum impulse section of about five stages instead of a Curtis ring on some of their large turbines. According to their latest designs, the impulse portion is now fitted with a two or three row Curtis wheel, the steam expanding in a high-pressure nozzle to about 45 lb. absolute. A single labyrinth balance piston takes up the axial thrust caused by this pressure. Reaction blading in the remainder of the turbines provides for the further expansion of the steam. The first turbine of this type was manufactured in the works of this firm in 1906 so that they claim to be the pioneers in the manufacture of this single-flow combined Curtis-Parsons unit.

22 The General Electric Company now manufacture a horizontal type Curtis turbine in all sizes up to 7000 kw., a section of which is shown in Fig. 3. This embodies all the essential features of the Curtis design. Compared with the vertical type, this design provides easier access to all working parts such as governor, bearings, valves, etc., and allows a better survey of the unit. The machine can also be dismantled and its internal parts examined with less trouble. The oiling problem is very simple compared with the vertical units, as there is no step bearing to provide for. It is possible that only horizontal units of all sizes will be built in the near

future. The A. E. G. also builds similar turbines up to 1000 kw. capacity.

23 Many manufacturers in Europe are now building a turbine of the type shown in Fig. 4. This consists of a Curtis high-pressure with Rateau or Zoelly low-pressure sections, the number of stages depending on the size of the machine, the speed and the steam conditions. The pressure and temperature in the casing are low, as a consequence of the expansion of the steam in the nozzles. The steam velocity is moderate in the low-pressure section. This ma-

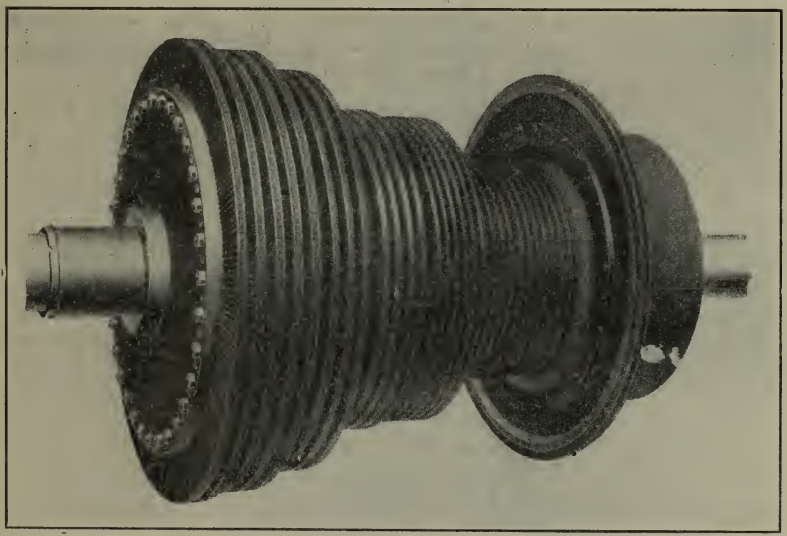


Fig. 5 Spindle of Curtis Drum-Impulse Turbine of Belliss & Morcom

chine is longer than the simple Curtis but shorter than the Rateau. Its builders claim improved efficiencies over either of the fundamental types. The type shown in Fig. 4 was worked out as a standard construction by the A. E. G. as early as 1904.

24 Belliss & Morcom, of Birmingham, England, have introduced a turbine, the spindle of which is shown in Fig. 5. It consists of a high-pressure Curtis stage with a low-pressure drum construction like Parsons, but fitted with impulse blading on the spindle and blades forming expansion parallel wall nozzles in the casing. All other impulse turbines employ diaphragms extending to the shaft. There are a greater number of rows in a Belliss turbine than in a corresponding Rateau turbine, and hence a lesser pressure drop through each set of guide blades. The net leakage loss may thus be

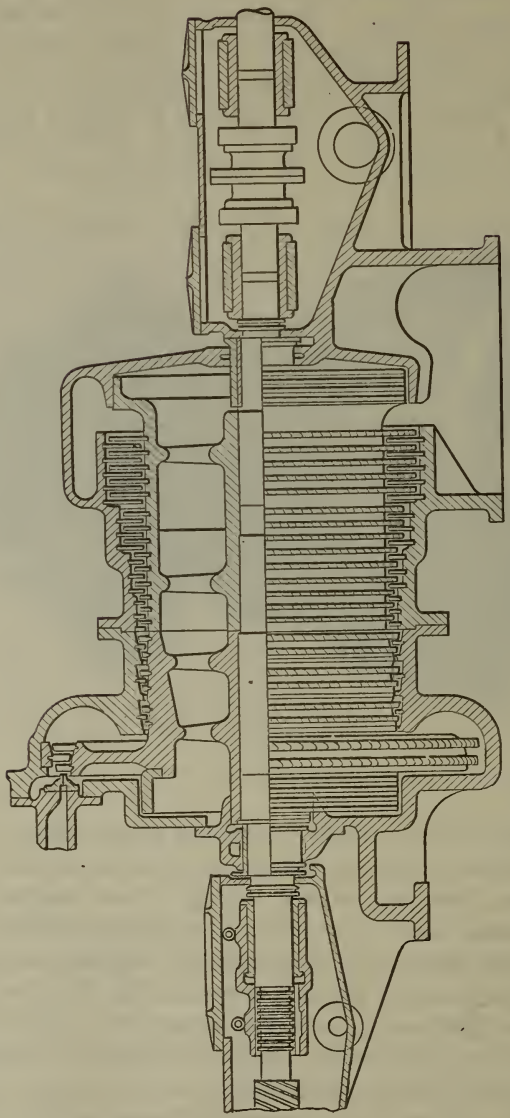


FIG. 6 SECTION OF CURTIS DRUM-IMPULSE PARSONS TURBINE

lower than the Rateau, even though the leakage area is greater. This turbine is essentially a Curtis-Parsons construction, with few rows of blades and with the expansion taking place in the stationary blades only.

25 A turbine, shown in Fig. 6, has recently been patented in England, which has some features of unusual interest. The high-pressure stage is Curtis, the intermediate section is of a drum impulse type with three stages, while the low-pressure portion is Parsons, which may be also double-flow. This combines features of all three fundamental types, and it will be interesting to see what results are obtained in practice by such an arrangement.

DISCUSSION OF TURBINE DETAILS

26 In the early days of steam turbine building it was difficult to secure suitable materials to withstand the stresses set up at high speeds of rotation. But as the demand for such materials increased, much study was given to the requirements for this service so that it has been possible through the use of more suitable material to increase very considerably the speeds of all sizes of steam turbines.

27 Higher steam velocities are possible with increased peripheral speed, and thus fewer rows of blades or stages are required. This results in a shorter and more rigid shaft construction, which is therefore less liable to vibration. Many builders, especially those of Parsons turbines, have found this construction to give an increase in steam economy over the slow-speed types, so that the more compact modern high-speed machine is more desirable than the older type.

BLADING

28 The requirements of a satisfactory blading material are that it shall withstand without deformation stresses due to centrifugal force, and temperature or pressure changes, it shall not cut out with high velocities of steam, and that it shall withstand the corrosive effects of moisture. Parsons turbines have used special bronzes and copper-nickel alloys. Steel blades have also been used in some cases. In these turbines the steam velocities are low and there is usually no cutting on this account. The principal problem with this blading is to manufacture it cheaply and secure it in such a manner that it will withstand all stresses to which it is subjected. This blading design is therefore a question of detail, and, as shown in Table 1, many ingenious schemes have been devised.

29 In Parsons turbines the blades are usually cut, punched or pressed into proper form from strips of drawn material. The original Parsons blading consisted of alternate blades and distance pieces placed in a slightly dovetailed slot and calked tight. Many European builders thread blades and distance pieces on holding wires before calking in. In this case the blading is made up in sections. Other builders of turbines of the Parsons type use Sankey's solid foundation ring held in place by a soft metal calking strip. Allis-Chalmers employ this well-known construction. As this form is usually all machine made, it is considered by many engineers to be safer than where dependence is placed on hand work, such as must ordinarily be done where each distance piece is calked separately.

30 There are many methods in vogue for spacing and reinforcing the ends of Parsons blading. The Westinghouse Machine Company use a comma-shaped wire threaded through the blades near their outer end and bent over between them. Similar schemes are used by manufacturers in Europe. As a rule, however, European builders follow the old Parsons method of silver soldering or brazing the blades to a holding wire near their outer ends. They generally thin down the tips of the blades to reduce weight and to avoid injurious effects to spindle or casing from accidental rubs. Some builders, such as Sulzers, do not thin off their blades or use shrouds, but make their spindles so rigid and well balanced that the blades can be made with a very heavy cross-section and hence need no support. Several manufacturers rivet the outer ends of their blades into channel-shaped shroud rings. This gives an especially stiff construction.

31 Advocates of the shrouded blading claim that it provides a labyrinth passage for the steam and thus reduces the leakage losses from row to row. It also holds the blades at the required angles. It has been noted, however, that with wet steam there is a tendency for the moisture to pit the casing opposite the edges of the shrouds and thus increase the clearances. This action has also been noticed with unshrouded Parsons blading. The shrouded blading is usually so stiff that serious damage is done if rubbing starts between the blading and the spindle or casing.

32 The blading of impulse turbines is of nickel steel, frequently with 25 per cent nickel, in the high-pressure section and special bronze in the other stages. Experience with this 25 per cent nickel steel blading material has not been entirely satisfactory, and several manufacturers are now using a low carbon steel alloy with just sufficient nickel to prevent corrosion, usually about 5 per cent. These

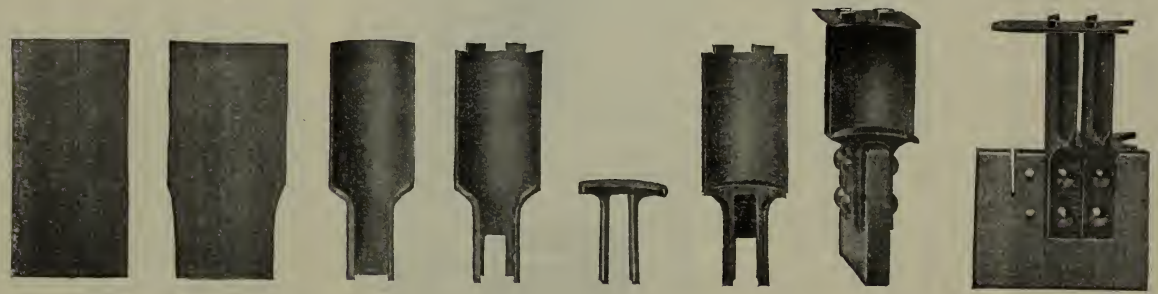


FIG. 7 STEPS OF MANUFACTURE OF BLADING FOR RATEAU SECTIONS OF BERGMANN TURBINE

blades are said to be stronger and less liable to fatigue of material. Special bronze and monel metals have also been successfully used. These impulse blades are stamped from sheets, drop forged or milled from solid bars with or without a wide base to act as a distance piece, or are made from extruded metal strips of the desired cross-section. Usually these blades are of crescent section, but some are formed of flat strip material and made of constant thickness over the width of the blade. The separate distance pieces are usually of the same material as the blades themselves. In general, all impulse blades are provided with shrouds to prevent vibration and also to provide an enclosed passageway for the steam at high velocities. As there

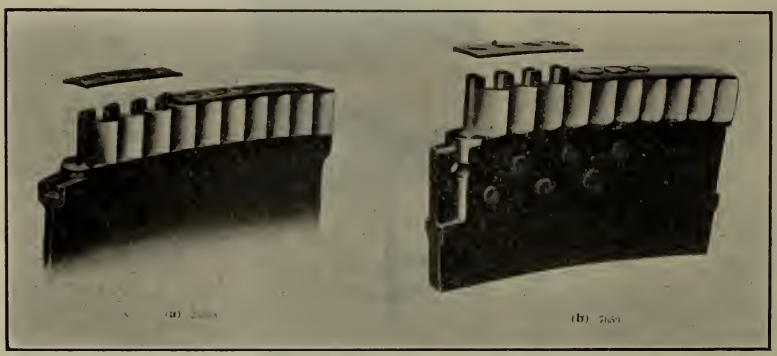


FIG. 8 TYPES OF BLADING USED BY BRITISH WESTINGHOUSE COMPANY

is no drop in pressure between the two sides of a row of moving blades, the clearance can be made large, both on the end and sides, so that there is little possibility of rubbing when in operation.

33 Impulse blading is usually held in place in dovetailed grooves or in tee-shaped slots, although some manufacturers form their blades with two legs which straddle the disks and are held firmly in place by rivets.

34 The first impulse turbines had blading in which the inlet and discharge angles were equal. Now almost all builders use blading on which the discharge side of the blade makes a sharper angle with the axis than the inlet side. This does not necessarily mean that the discharge area of the blades is smaller than the inlet area, for the blade is usually lengthened radially on the discharge side. Thus both inlet and discharge areas are made equal. The sharper angle of discharge reduces slightly the relative velocity of exhaust from the moving blade. On turbines of the Curtis type there is usually

only a small difference between the inlet and outlet angles of the first row of moving blades, but on the second row of moving blades in the stage, the entrance and exit angles often differ by as much as 15 deg.

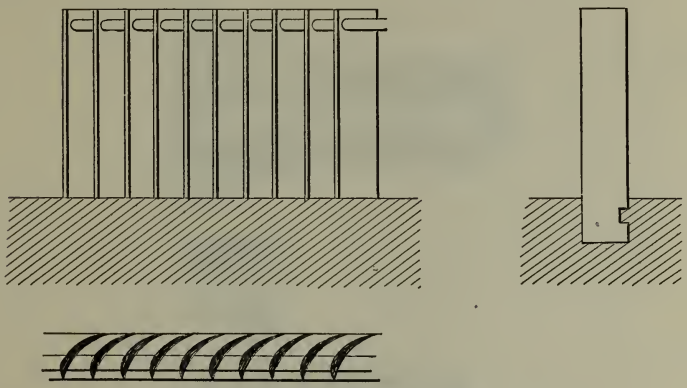


Fig. 9 Brown Boveri's Reaction Blading

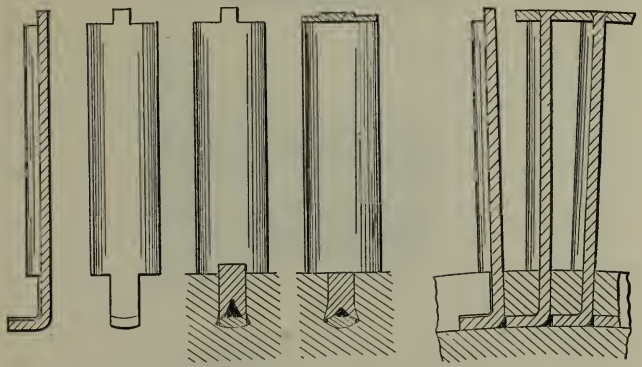


Fig. 10 Belliss & Morcom Impulse Blading

35 Some impulse turbine designs are such that there must evidently be some such reaction effect in the moving blades as is obtained in Parsons turbines, though not of sufficient amount to cause any noticeable end thrust.

36 Some recent interesting developments in blading are shown in Figs. 7-11. Fig. 7 shows the various steps in the manufacture of the thin sheet metal blading used by Bergmann. The blading of the Rateau disks of British Westinghouse turbines is clearly illustrated in Fig. 8. Brown Boveri's form of Parsons blading is illustrated in

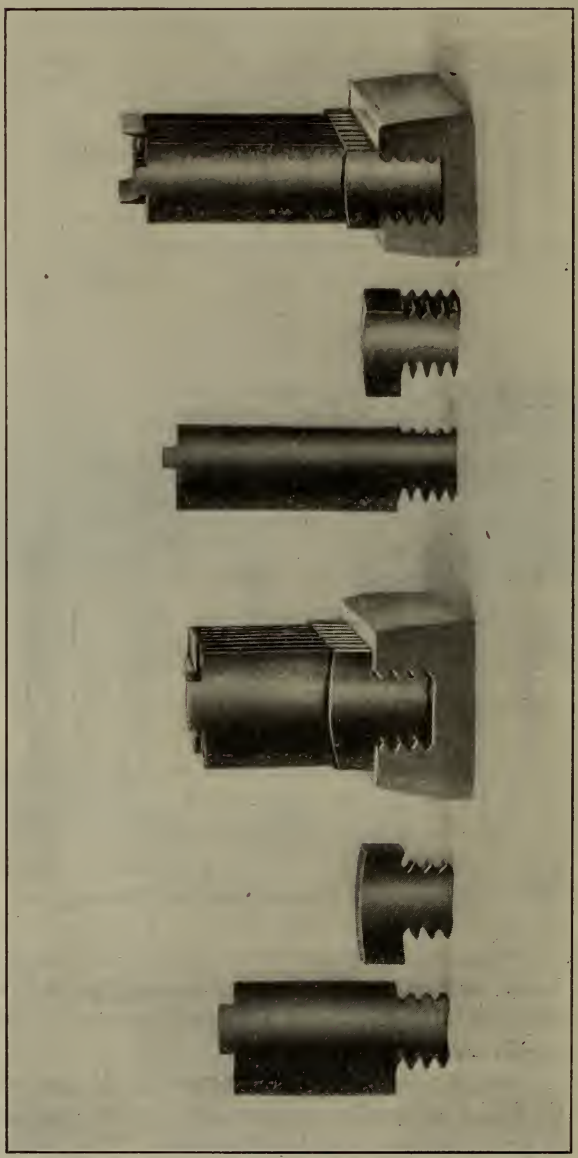


FIG. 11 FRANCO TOSI'S BLADING. IMPULSE BLADES ON LEFT, REACTION BLADES ON RIGHT

Fig. 9. Belliss & Morcom use a unique form of blading on the drum impulse section. This blading is shown in Fig. 10. Franco Tosi cuts projections like threads in his blade grooves and mills corresponding projections on the base of the blades, as in Fig. 11. This is a very satisfactory but expensive form of blading.

37 The number of rows of blades in any given type depends entirely on the size and speed of the unit, the steam conditions under which the turbine will operate and the heat drop per row or stage assumed in the design. Each designer selects such conditions as his experience and judgment lead him to consider the most satisfactory. It is thus hard to draw any general conclusions as to the most desirable number of rows for any class of turbine. In general, it may be said that the Parsons turbines require from 40 to 80 sets of moving and stationary blades, the Curtis from 4 to 8 stages, the Rateau from 12 to 25 stages, the Zoelly from 8 to 20 stages, while the Curtis-Parsons and Curtis-Rateau combinations of course require several rows less than the fundamental types.

NOZZLES

38 The high-pressure nozzles in impulse or combination turbines are made of bronze or nickel steel highly polished inside and placed either in the upper end of the casing itself, or, more preferably, in a separate steel casting which bolts upon the casing. In the latter construction the casing itself is never subjected to the high temperature or pressure of the entering steam.

39 The nozzles in the diaphragms between stages are usually made of nickel steel or other special steel, bent to the proper form and cast in place in the diaphragm body. Some manufacturers use brass nozzles in place of steel made up in sections and riveted or bolted in place. All these passages must be smooth and preferably very highly polished to reduce friction losses. After the first stage all nozzles have parallel walls on the discharge side.

BEARINGS AND LUBRICATION

40 Practice varies widely with regard to the design and construction of journals and bearings. Builders of impulse turbines invariably use cast-iron bearing shells provided with spherical self-aligning pads and lined with white metal. These are being used to an increasing extent on Parsons turbines and without exception on all low-speed units. Some manufacturers still retain the original Parsons form of bronze bearing shell with concentric rings on the

outside, separated from each other by oil films. It has been claimed for this type of bearing that the oil films dampen any slight variation of the spindle and thus provide a machine running more quietly. Experience has shown that this is not always the case and that such a bearing is often a real source of danger when the added clearance of the spindle due to play between the rings is taken into consideration. This construction is also much more expensive than the white metal bearings.

41 In Europe many turbines are run with a minimum amount of oil and with oil leaving the bearings at a temperature of 190 deg. fahr. This practice is based on the argument that such a system of lubrication requires the least expenditure of power for oil circulation and in friction losses. However, practice seems to be tending towards flooded lubrication, in which a great quantity of oil at a temperature of about 100 deg. fahr. is forced through the bearings by a pump of the rotary, centrifugal or gear type, driven from the main shaft of the turbine. The oil pressure at the bearings varies from 3 to 20 lb. per sq. in. The life of the oil is much longer in this system than with very hot oil and any wear on the bearings themselves is absolutely prevented. Occasionally the bearing shells are water-cooled, but this practice should be discouraged. Cooling water can be used much more satisfactorily outside of small oil pipes in properly designed coolers. In case the water is dirty or full of scale-forming impurities, these can be more readily removed from an oil cooler than from the interior of a bearing shell.

42 Flooded lubrication has enabled manufacturers to cut down the length of their bearings and thus reduce the total length of their turbines. The increased pressures per unit area on the bearings have not introduced any difficulties, so that pressures of 80 to 100 lb. per sq. in. at a surface speed of 60 ft. per second are common practice. The best results are usually obtained with a temperature of about 125 deg. fahr. as the oil leaves the bearings.

SPINDLE CONSTRUCTION

43 Parsons turbines in America are usually built with a hollow quill into which the journal ends are forced and fastened by shrink links or bolts. The high-pressure blading is placed in grooves in one end of the quill itself. The intermediate and low-pressure blades are usually carried on cast or forged-steel rings which are afterwards forced and keyed upon the central quill. In Europe excellent hollow steel forgings can be obtained very readily, and hence the spindles

of Parsons and other drum type turbines are usually made up of one forging with the journal shafts fastened into the ends.

44 The shafts of impulse turbines are usually in one piece and carry the blade disks, which are high-grade steel or nickel steel forgings or castings. These disks are fitted and keyed on the shaft and held in place by shrink links or lock nuts.

45 It is quite general practice now to design the shafts so that the normal speed of the turbine will be very considerably below the critical speed due to any slight unbalancing of the mass that may be present. This removes the dangerous vibrations often experienced when passing through critical speeds and permits closer clearances to be used on Parsons blading and in the labyrinth passages in the diaphragms between the stages of impulse turbines.

PACKING GLANDS

46 A small impeller supplied with water is provided on all American built Parsons turbines to form an air seal at the shaft glands. European builders prefer to use labyrinth packings with live or throttled steam as an air seal. The objection raised by foreign builders to the water packing is that it takes too much power to drive the impeller, and that it provides a condensing surface for the steam. The amount of water required in a well-designed gland is very small, and there is no great circulation. Thus the water can have only a small effect as a condensing medium. Usually the steam directly inside the casing is under vacuum and then condensation would not be objectionable. It requires considerable steam to pack the labyrinth type of gland, and this loss often exceeds that due to the power required to drive the water impellers.

47 Impulse turbines use carbon rings at the high-pressure gland, especially when superheated steam is used. The leakage past the first rings is carried through a passage and pipe to the low-pressure glands to act as a vacuum seal. The labyrinth packing in the diaphragms is usually of bronze in the high-pressure stages and frequently of white metal in the low-pressure section. The low-pressure shaft glands are made either with carbon or labyrinth packing, sealed by steam.

THRUST BEARINGS

48 Thrust bearings are now provided on all types of turbines. These serve to adjust the position of the spindle and to take up any end thrust present. The end thrust in well designed turbines is

usually of small amount and seldom causes trouble. These bearings are usually flooded with oil and are often made with bronze rings which can easily be replaced in case of damage.

49 Several builders of Curtis-Parsons turbines have substituted an oil thrust piston for the steam balance piston of the usual form. This piston is placed on the governor end of the shaft outside of the casing proper. The thrust is taken up by oil which is supplied under pressure and which can escape only through sets of labyrinth baffles on the piston and on the surrounding chamber. The piston adjusts itself for the amount of thrust present by a small axial movement of the shaft, thus opening or closing the discharge area for the oil through the labyrinth baffles, and automatically varies the oil pressure on the face of the piston. This arrangement does away with the loss of steam through balance pistons and should improve the turbine efficiency, though a small amount of power is required to pump the oil.

GOVERNING DEVICES

50 The speed of Parsons turbines is usually controlled by a centrifugal governor of the Hartung or similar type which regulates the position of a balanced poppet valve through the medium of a steam or oil relay. The latter type is coming into more general use on account of its many advantages over the steam relay. The steam is throttled either at constant pressure or by a pulsating action.

51 Impulse turbines and turbines employing a Curtis high-pressure ring usually govern by means of Hartung type governors and oil relays. The speed is sometimes controlled by simple throttling of the steam, which practice is common in Europe. In this case, additional nozzles can be opened or closed by hand as required. This system would be unsuitable with violently fluctuating loads. Other types employ both throttling and automatic nozzle regulation, while, again, many turbines, particularly of the Curtis type, are built for nozzle governing alone. In Europe there is a difference of opinion as to the most economical method of governing. Mr. H. Zoelly claims that he gets the best results on his type of turbine by throttle governing. It is generally admitted, however, that impulse turbines give the best results by controlling the number of nozzles that are open at any load. With such control the pressure before the nozzles in service is always the normal pressure for which these are designed. The General Electric Company govern their turbines above 300 kw. by means of an oil relay system which oper-

ates through a piston and camshaft to open or close nozzles as required by load conditions. Melms and Pfenninger employ an interesting governing device, shown in Fig. 12. The speed regulation

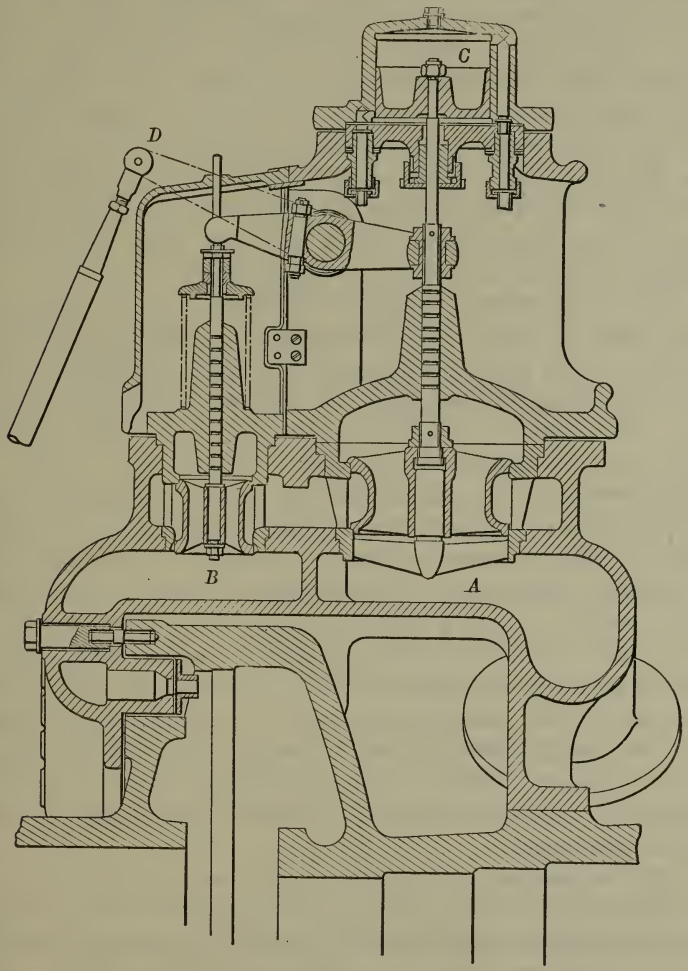


Fig. 12 Melms and Pfenninger's Governor Device

is taken care of by a shaft governor of the Hartung type operating, through oil relays, a "servomotor" or oil operated piston C, which is directly connected to the main valve A, and also indirectly connected through levers and a secondary shaft to three balanced valves

controlling the steam supply to the three sets of nozzles as shown in Fig. 12, where B is one of the three auxiliary valves, and D the lever controlling opening of valves, operated by governor through an oil relay valve. The main valve thus acts largely as a safety device. Each of the secondary valves is designed to take care of only a certain predetermined portion of the load. These valves which open downwards are counterbalanced by springs and are acted upon by levers mounted on the secondary shaft in such a manner that the valves are successively opened or closed according to the change of load. The secondary shaft, as previously noted, is under direct control of the governor through the servomotor and its connecting lever. All nozzles, casings, valves, etc., are made of cast steel. This device is said to give very close regulation.

52 The Westinghouse Machine Company use a vibrating oil relay system on many of their machines. Recently they have adopted on some sizes a very powerful governor which is direct-connected to the governing valve and operates it without the use of relays. This system was adopted by some of the early European builders, but was abandoned in favor of the oil relay system.

53 Sulzers have recently installed turbines in which the conventional centrifugal force governor has been replaced by a hydraulic governor. This governor consists of a simple centrifugal pump, geared to the main shaft and delivering oil under pressure into a chamber beneath a spring loaded piston. The pressure under this piston is thus dependent on the speed of the main turbine and the piston's position in its cylinder will vary accordingly. The piston is connected to the usual balanced valve of an oil relay which controls the oil supply from the main oil pump to the throttle valve in the usual manner. There are no mechanical parts to wear in this arrangement. The apparatus is extremely simple and has many distinct advantages.

54 All turbines are now provided with a small overspeed governor, usually placed at the outside end of the shaft. This operates at a determined percentage over speed and closes the main or secondary steam valve either by means of a steam or oil relay or by a falling weight through a system of levers and springs. The oil relay system has the advantage that the valve shuts immediately should the oil supply fail for any reason.

55 Parsons turbines are usually provided with a secondary overload valve which automatically admits live steam to the second diameter of blades. Impulse turbines have additional sets of nozzles

with valves which may be opened automatically, or by hand in case of overloads.

CASINGS

56 Practice varies widely in regard to the construction of turbine casings. These are generally made of cast iron, though some European builders make the high-pressure front end of cast steel. Parsons turbines are built with the top and bottom halves single castings or made in sections. Some of the older designs of impulse turbines used solid diaphragm plates placed on the shaft between the disks. The clumsiness of handling and the difficulty of making repairs with this construction has forced most builders to make these in halves and to fasten them to the top and bottom portions of the casing.

57 Cylinder casings are now made of symmetrical design and without any deep metal webs or ribs as stiffeners on the outside. Equalizing pipes with provision for expansion take the place of passages formerly cast in the casings themselves. Strains due to unequal temperatures must be avoided in all portions.

BED-PLATES

58 Several European builders make a practice of filling the hollow portions of their turbine and generator bed-plates with concrete after erection. This adds more mass to the turbine unit and is said to dampen any slight vibration that may be present.

TURBO-GENERATORS

59 A discussion of the design of turbo-generators is beyond the limits of this paper. In general, European builders of electrical machinery allow less overload capacity than is usual on American machines, but give better guarantees of efficiency and regulation.

60 Practice varies widely in regard to the normal rating and maximum capacities of different turbines. Some European builders guarantee their generators to carry normal full load without undue heating only for two to six hours. Others follow the practice which has been introduced in America of rating their turbines at the maximum load they will carry continuously. Several builders still offer turbines which will carry as high as 25 per cent overload continuously.

61 Such conditions are very confusing to purchasers. The time seems opportune for this Society to fix a standard for the rating of steam turbines and to define the overload capacity which may be expected of this type of engine.

COMMERCIAL CONSIDERATIONS

62 When steam turbines are to be installed at high altitudes, as in many locations in the West, the effect of altitude on economy frequently comes up. It can be easily shown that with the same boiler pressure and the same absolute pressure in the condenser, the steam consumption of a turbine at 5000 ft. elevation should exceed that of a similar turbine at sea level by less than 1 per cent.

63 Each turbine is designed to operate at maximum efficiency at some given vacuum. Owing to uncertainties in design factors for losses and to slight inaccuracies in construction, the most efficient vacuum may vary somewhat from that for which the turbine was designed. Theoretically, the steam consumption should decrease as the vacuum increases, but this rate of decrease will vary for each type of turbine, depending on the blade areas and steam velocities of the low-pressure section. It is, therefore, essential that this rate of change be determined for each individual turbine by actual test.

64 In the purchase of steam turbines, operating efficiency and costs should be considered together with first cost, and the machine selected should be the one on which the ultimate operating and fixed charges are a minimum. This practice is followed in Europe. In America the bargain-counter idea unfortunately possesses many engineers, and manufacturers have frequently been forced to sacrifice efficiency in order to meet competition.

65 In Table 1 some data are presented in tabulated form regarding various types of steam turbines. This information was gathered during personal visits to the various works and also by correspondence. Designs of steam turbine details are constantly changing, so that many of the items in this table may not correctly represent the latest practice of the various builders.

RESULTS OF TESTS

66 The efficiency of a steam turbine may be expressed in terms of pounds of steam per kw-hr., as an efficiency ratio or as the B.t.u. required per kw-hr.

67 The steam consumption is dependent on the initial steam pressure, its temperature or quality, and the condenser pressure. These factors vary in almost every test and the effect of a variation in each is not the same for all classes of turbines. Hence, different turbines cannot usually be compared satisfactorily on the basis of their steam consumption alone.

68 If the steam could expand freely to exhaust pressure in a turbine without radiation, friction, eddy or windage losses, its expansion would be adiabatic and on the Rankine cycle. The "efficiency ratio" expresses the proportion of the heat actually turned into work to that available from such an adiabatic expansion. In other words, it expresses the efficiency of the actual turbine as compared to the ideal turbine, and is independent of the type of turbine.

69 The B.t.u. per kw-hr. is figured above the heat of the liquid at exhaust pressure. This is not a satisfactory standard by which to compare results, for it is largely dependent on conditions beyond the control of the turbine builder. For instance, if the plant does not contain superheaters, the B.t.u. per kw-hr. will be high. The same will be true of a plant which has a warm cooling-water supply for condensers and consequently carries low vacuum. Yet the turbines may be designed to give a high efficiency ratio under these conditions. In fact, they may be able to utilize the available heat more efficiently than the turbines in another plant with both high superheat and high vacuum.

70 This can be seen in Table 2, in which recent turbine tests have been tabulated. The Brown-Boveri turbine at the Dunstan power plant uses 14,980 B.t.u. per kw-hr. with an efficiency ratio of 68.8 per cent. Yet the Westinghouse City Electric with a lower steam pressure, lower superheat and lower vacuum, has an efficiency ratio of 68.9 per cent, though using 16,925 b.t.u. per kw-hr. The Erste Brünner Vienna turbine requires 16,460 B.t.u. per kw-hr. with 71.8 per cent efficiency ratio. It is therefore apparent that the efficiency ratio alone will express in the best manner the degree to which the designer has approached ideal results in his turbine.

71 The test results in Table 2 were grouped in order to analyze the relative merits of the different types of turbines on the basis of efficiency ratios. The Curtis-Parsons machines built by Erste Brünner hold first place in the list, but are followed closely by others of the same type built by Brown Boveri and Westinghouse Machine Company. The Parsons turbines built by Allis-Chalmers and Brown Boveri also show high efficiencies. The second class in the order of efficiency includes turbines of the Curtis-Rateau and Curtis-Zoelly types, among which the turbines of the A. E. G. and British Westinghouse Company show remarkably good results. The next group includes simple Zoelly and Rateau turbines. The last group comprises straight Curtis types.

TABLE 1 DATA ON STEAM TURBINE CONSTRUCTION

Manufacturer	Type of Turbine	R.P.M. of Various Sizes	No. of Stages.	Blade Speed in Ft. per Sec.	Steam Velocities, Ft. per Sec.	Blade Material
C. A. Parsons & Co.	Parsons & Curtis-Parsons	.....	Usually about 70	270 on low-pressure blades	540	Special bronze and delta metal.
Westinghouse Mach. Co.	Parsons & Curtis-Parsons	Up to 3000 kw., 3600; 3000-5000 kw., 1800; over 5000 kw. (750 or 720)	.....	Curtis blades 600; Parsons blades 350	Curtis 2000; Parsons 600	Special bronze and soft steel; Curtis blades bronze
Willans & Robinson.	Curtis-Parsons	.....	1 Curtis; 12-20 rows Parsons on 1 diameter of drum	.....	.....	Bronze and copper-nickel blades
Brown Boveri & Co., Richardsons Westgarth & Co.	Parsons & Curtis-Parsons	Up to 3500 kw., 3000; 3000-7500 kw., 1500	.....	Curtis blades 460; Parsons blades 380	.....	Curtis blades of special alloy; Parsons blades of bronze
Brush Electrical Engrg. Co.	Curtis-Parsons	.....	.....	.....	.....	Parsons blades bronze alloy
Erste Brünner M. F. G.	Curtis-Parsons	Up to 3500 kw., 3000; 1500-4000 kw., 1500; 4000 kw., up, 960	1 Curtis; remainder Parsons on 2 diameters of drums	.....	.....	Both Parsons and Curtis blades of special bronze
Franco Tosi.	Curtis-Parsons	Up to 2500 kw., 3000; 2500-8000 kw., 1500; 8000-12,000 kw., 1000	1 Curtis; remainder Parsons on 1 diameter of drum	.....	Curtis 2130; Parsons 230-460	Parsons and Curtis blades of special bronze
Gebrüder Sulzer.	Curtis-Parsons	.....	.....	Curtis blades 360; Parsons blades 310	.....	Curtis blades all of special steel; Parsons blades of special bronze
Melms & Pfenninger, Breitfeld, Danek & Co.	Rateau-Parsons & Curtis-Parsons	Up to 4000 kw., 3000; 4000-9000 kw., 1500	.....	Up to 575	Impulse section up to 2200; Parsons up to 900	Rateau blades of special bronze; Parsons blades also special bronze
Allis-Chalmers Co.	Parsons	Up to 2000 kw., 3600; 2000-5000 kw., 1800	.....	Up to 370	400-800	Copper-nickel alloy
Allgemeine Elektricitäts Gesellschaft.	Curtis-Rateau	Up to 5000 kw., 3000; 5000-12,000 kw., 1500; 12,000 kw. and over. 1000	1 Curtis; 4-12 Rateau	Up to 600	Curtis 2600; Rateau about 1200	Bronze alloy or nickel-steel

Maschinenfabrik Augsburg Nurnberg.	Curtis-Zoelly	Up to 6000 kw., 3000; 6000-10,000 kw., 1500; 10,000-15,000 kw., 1000	1 Curtis; 6-8 Zoelly	Up to 650	Curtis 2500; Zoelly about 1300	Nickel-steel highly polished
British Westinghouse Co.	Curtis-Rateau	Up to 1500 kw., 3000; 1500-5000 kw., 1500	1 Curtis; 6-12 Rateau	300-500	Curtis 1500; Rateau 800	5% nickel-steel
Bergmann-Elektricitäts-Werke.	Curtis-Rateau	Up to 6000 kw., 3000; 7000-12,000 kw., 1500; over 12,000 kw., 1000	1 Curtis; 8 Rateau	410-550	Curtis 2600; Rateau 1400	25% Wolfram steel
Belliss & Morcom.	Curtis-Drum Impulse	Up to 1500 kw., 1800; use lower speeds than most builders	1 Curtis; 25-30 rows of impulse blades on drum	Not exceeding 400	.....	Curtis blades of steel; Impulse blades of phosphor bronze
Escher Wyss & Co.	Zoelly	Up to 7000 kw., 3000; 7000-12,000 kw., 1500; over 12,000 kw., 1000	3000 r.p.m., 10 stages; 1500, 16 stages; 1000, 20 stages	.....	About 1400	5% nickel-steel
Maschinenfabrik Oerlikon.	Rateau	Up to 4000 kw., 3000; 4000-7500 kw., 1500; over 7500 kw., 750	10-16 stages with 2 or more mean diameters of blades	320-520	1310-1700	Nickel-steel
Sächsische Maschinenfabrik.	Zoelly	Up to 4000 kw., 3000; up to 7000 kw., 1500; for outputs over 7000 kw., 1000	1 Curtis; 6-12 single rim	190-200	600-800 in front of Curtis wheel; 300-400 in front of pressure wheels	Nickel-steel
Frazer & Chalmers. Pokorny & Wittekind.	Rateau	.....	Are reducing number of stages on all turbines	Have increased blade speeds recently	.....	Nickel-steel
	Curtis-Rateau & Rateau	.....	.....	500-600	2300 admission	.....
General Electric Co.	Curtis	Up to 1400 kw., 3600; 1400-7000 kw., 1800; 7000-20,000 kw. (750 or 720)	4-7	460-500	About 1200 average	Special bronze extruded metal; Monel metal and steel used on some low-pressure blades
British Thomson Houston Co.	Curtis	Up to 1000 kw., 3600; 1000-5000 kw., 1800; up to 2000 kw., 3000; 2000-7500 kw., 1500	Same construction as General Electric Co.	600 as maximum	About 1800 from nozzles, 1100 from diaphragms	Special bronze and steel

TABLE 1—*Continued*

Manufacturer	Blade Form	Method of Blade Manufacture and Insertion
C. A. Parsons & Co.	Parsons form with very thick rounded inlet, blades thinned off at tips	Alternate distance pieces held by calking, ends of blades stiffened by silver soldering to a brass strip; Curtis blades shrouded
Westinghouse Mach. Co.	Blades crescent with heavy central portion and sharp inlet edge. dovetail base on impulse	Alternate distance pieces held by calking, ends stiffened by comma wire threaded through blades and bent over
Willans & Robinson.	Parsons rounded inlet, stamped to fit Sankey foundation ring and channel shroud; usual Curtis blades on impulse	Parsons sections made up by machine and held in place by calking in soft metal strip, final riveting of shroud completed last
Brown Boveri & Co., Richardsons Westgarth & Co.	Parsons blades not thinned off, usual form, base of blade notched to fit corresponding lateral projection in the grooves, blades not calked	Curtis blades in dovetail slot, no sharp corners on base and shrouded; Parsons blades threaded on wire and silver soldered
Brush Electrical Engrg. Co.	Usual Parsons form with calked distance pieces in dovetailed groove; also use grooved base of blades fitting corresponding grooves in shaft like teeth in a bolt	Tips of blades thinned off, blades silver soldered to wire strip
Erste Brünner M. F. G.	Curtis blades have special dovetail base and are separate from their distance pieces; Parsons blades usual form with distance pieces in dovetail groove	Curtis blades shrouded; Parsons blades thinned at tips and silver soldered to wire on side
Franco Tosi.	Blades and distance pieces milled at base, with thread-like projections to fit corresponding grooves in spindle	Curtis blades shrouded; Parsons blades also have channel shroud-riveted over after blades are in place
Gebrüder Sulzer.	Parsons blades of very heavy cross-section with distance piece drop-forged and fitted with a projecting nose on the concave side which fits into a milled slot on the convex side of the blades	Parsons blades not thinned at ends or reinforced in any way; Curtis blades milled to fit dovetail slot
Melms & Pfenninger, Breitfeld, Danek & Co.	Impulse blades have slot in side to fit projection in groove and held in place by soft metal calking strip; Parsons blades have special base to fit foundation ring and with special shroud on end	Parsons blade rings made up in halves and fitted to machine, sections held in place by soft metal calking strip, angle of blades, 20 to 45 deg.

Allis-Chalmers Co.	Parsons section with more defined inlet edge, blading uses Sankey foundation ring and shroud, blading made in sections outside turbine	Sections held in dovetailed slots by soft metal calking strips, shrouds channel-shaped
Allgemeine Elektricitäts Gesellschaft.	Blade crescent shaped but of lighter section toward exhaust end, shrouded in groups	Blades cut from solid bar and set in dovetailed grooves with separate distance pieces; nickel-steel blades milled
Maschinenfabrik Augsburg Nurnberg.	First Curtis rows have blades of heavy section, remaining rows are thin in section, blades are held in a ⊥ slot	Blades mostly drop-forged, all rows shrouded, outlet and inlet angles vary considerably
British Westinghouse Co.	Curtis blades have ⊥ base to fit corresponding groove; Rateau blades have 2 projections at base straddling disk and rivets rolled in place	Blade and distance pieces milled from solid steel bars
Bergmann-Elektricitäts-Werke.	Blades stamped from sheet metal with projections at base straddling disk and riveted to it, blades of uniform thickness across section	Distance pieces drop-forged of special bronze and riveted with blades to disk, all blades shrouded
Belliss & Morcom.	Curtis blades, usual form; on impulse section blades are of uniform thickness over section and shrouded	Blades made with strip at base bent in form of L, distance piece formed with opening at base, the 2 projections spread over base of blade and fill dovetailed slot when calked down
Escher Wyss & Co.	Blades stamped from sheet metal with ⊥ base, blades sometimes thinned off toward tips, distance piece stands well above surface of shaft	Blades milled with sharp edges and highly polished after stamping, distance pieces milled from solid bar, only thin shroud ring used
Maschinenfabrik Oerlikon.	Blades punched to shape from drawn metal strips, base is ⊥ shaped, simple band shroud ring used	Distance pieces milled, slot closed by special locking piece
Sächsische Maschinenfabrik.	Zoelly blades thinned off toward tips on low-pressure end, base ⊥ shaped with rounded corners	Blades and distance pieces milled from solid bar, channel-shaped shroud
Frazer & Chalmers.	Blades with forked base riveted to disks	Blades milled from solid bar, no shroud ring, but head milled on blades like De Laval
Pokorny & Wittekind.	High-pressure blades shrouded, low-pressure blades unshrouded on some units	.....
General Electric Co.	Blades milled with dovetail base and have shroud ring, distance pieces also milled	Blades put in place in special blading machine
British Thomson Houston Co.	Same construction as General Electric Co.	Same construction as General Electric Co.

TABLE 1—*Continued*

Manufacturer	Nozzles and Diaphragms	Shaft and Spindle Construction	Form and Material of Bearings
C. A. Parsons & Co.	.....	Spindles usually a single hollow forging, shaft ends fastened in	Parsons concentric bronze sleeve bearings in self-aligning cast-iron casings, large sizes white metal lined
Westinghouse Mach. Co.	.....	Shaft, hollow quill with blade rings forced on, ends held in by special patented fastening	Parsons concentric sleeve bearing on small units, cast-iron self-aligning bearings, white metal lined on large units
Willans & Robinson.	.....	Disk, solid forging; drum, hollow rolled forging; ends fastened in by locked bolts; Fullager balance piston used	Cast-iron, white metal lined, self-aligning
Brown Boveri & Co., Richardsons Westgarth & Co.	Nozzles made in 3 parts of steel or bronze and all machined	Disk, solid forging forced and fastened on hollow forged drum spindle; shaft ends locked into drum; Fullager balance piston used	Parsons concentric bearings on high speeds, cast-iron, white metal lined, self-adjusting bearings on low speeds
Brush Electrical Engrg. Co.	.....	Disk, solid and fastened on solid or hollow forged drum spindle	Cast-iron, white metal lined, self-adjusting bearings on all sizes
Erste Brünner M. F. G.	Nozzles of special bronze alloy	Small sizes, nickel-steel, solid rotors; large sizes, hollow forged high grade steel; Curtis ring bolted to spindle	Parsons concentric sleeve bearings on small units; on large units cast-iron white metal lined self-aligning bearings
Franco Tosi.	Nozzles of steel or bronze, angles of nozzles 20 to 25 deg.	Forged steel shaft with spindle rings and impulse disk with balance piston forced on and held by bolts	Cast-iron, white metal lined, self-aligning bearings
Gebrüder Sulzer.	Nozzles of forged steel, cut out from solid	Spindle, hollow forging with shaft ends and impulse disk bolted on	Bearings are steel castings lined with white metal
Melms & Pfenninger, Breitfeld, Danek & Co.	Nickel-steel guide blades in Rateau section with special labyrinth packing. angle of nozzles 18 deg. to 22 deg.	Spindle, hollow forging of nickel-steel, impulse disk and balance piston fastened on; shaft ends forced in	Parsons concentric sleeve bearings on small units, large units have cast-iron, white metal lined, self-aligning bearings
Allis-Chalmers Co.	.....	Hollow forged quill with blade rings of cast steel forced on; shaft ends forced into quill and held by bolts and buts: Fullager balance piston	Cast-iron, white metal lined, self-adjusting bearings

Allgemeine Elektricitäts Gesellschaft.	Nozzles of bronze or nickel-steel, guide blades in diaphragms of nickel-steel cast in place	Solid shaft with disks forced on; disks of steel tapering toward the blades	Cast-iron, white metal lined bearings, oil-cooled; only 3 bearings; small governor bearing carried by front housing
Maschinenfabrik Augsburg, Nurnberg.	Nozzles of nickel-steel milled from solid, guide blades in diaphragms of highly polished sheet nickel-steel	Short stiff solid shaft running below critical speed disks of soft steel forced on shaft and locked; disks taper toward blade ends	Cast-iron, white metal lined, self-adjusting bearings; only 3 bearings used; governor bearing very small
British Westinghouse Co.	Nozzles of special steel, guide blades of special steel cast in place in diaphragms	Short, solid, stiff shaft stepped from center to receive disks which are of forged steel tapering from center to periphery	Cast-iron, white metal lined, self-adjusting bearings
Bergmann-Elektricitäts-Werke.	Nickel-steel nozzles in 5-7 groups, nickel-steel guide blades cast in place in diaphragms	Short, solid shafts as shafts at 3000 r.p.m. and over, run under critical speed; disks forced on shaft and locked	Cast-iron, white metal lined, self-adjusting oil-cooled bearings
Belliss & Morcom.	Nozzles, steel or bronze, impulse guide blades of brass made in segments	Forged steel drum fastened to through solid shaft; drum carries impure disk and balance piston; heavy rigid construction.	Cast-iron, white metal lined, self-adjusting bearings
Escher Wyss & Co.	All nozzles and guide blades of nickel-steel cast in place and highly polished	Solid shaft with disks supported on T ring	Cast-iron, white metal lined
Maschinenfabrik Oerlikon.	Nickel-steel guide blades all cast in	Nickel-steel shafts made heavy in the center; disks of forged soft steel forced on and locked to shaft	Cast-iron, white metal lined, self-aligning bearings, water-cooled in the larger sizes
Sächsische Maschinenfabrik.	Sheet-steel guide blades cast in place	Solid shaft with nickel-steel disks, highly polished, forced on	Cast-iron, white metal lined, self-adjusting, water-cooled bearings
Frazer & Chalmers.	Diaphragm guide blades cast in place	Solid shaft with disks forced on	Cast-iron, white metal lined, self-adjusting bearings
Pokorny & Wittekind.	Angle of nozzles 20 deg.	Solid shaft with disks forced on	Cast-iron, white metal lined bearings
General Electric Co.	Inlet nozzles of special bronze, steel guide blades in diaphragms cast in place and polished	Shaft, solid forgings; disks, cast steel forced on shaft	Cast-iron, white metal lined, self-adjusting bearings with copper cooling pipes cast in babbitt metal lining
British Thomson Houston Co.	Same construction as General Electric Co.	Shaft and disks of steel	.....

TABLE 1—*Continued*

Manufacturer	Surface Speed of Journals, Ft. per Sec.	Bearing Pressure, Lb. per Sq. In.	Type of Oil Pump	Governor and Method of Governing
C. A. Parsons & Co.	35-50	40-50	.....	Steam relay system, steam admitted in puffs, not continuous throttling
Westinghouse Mach. Co.	40-50	50-70	Rotary oil pump with sliding gate	Direct throttling from governor, sometimes pulsating; also pulsating oil relay
Willans & Robinson.	45-60	50-70	Both reciprocating and rotary types used	Direct throttling from governor
Brown Boveri & Co., Richardsons Westgarth & Co.	.....	.....	Rotary pump	Hartung type with oil relay, also slight pulsation to valve; valves on nozzles open automatically
Brush Electrical Engrg. Co.	.....	.....	Rotary pump	Steam relay system, pulsating action
Erste Brünner M. F. G.	50	40-50	Double rotary valveless pump	Hartung type governor, steam relay and piston valve on main inlet; pulsating governing
Franco Tosi.	.....	55-90	Rotary gear pump	Hartung type governor, oil relay system on main valve, automatic oil relay nozzle governing
Gebrüder Sulzer.	.....	.....	Triple gear pump	Automatic hydraulic regulator, centrifugal pump on oil only; throttling of supply; oil relay on all gear; no mechanical governor
Melms & Pfenninger, Breitfeld, Danek & Co.	40-80	35-85	Eccentric rotary pump	Hartung type oil relay, throttling governor
Allis-Chalmers Co.	50-60	60-140	Rotary lobe pump, auxiliary duplex pump for starting	Hartung type oil relay, throttling governor
Allgemeine Elektricitäts Gesellschaft.	50-70	100-125	Rotary gear	Special type centrifugal governor operating oil relay, governor throttling steam; also automatic nozzle control

Maschinenfabrik Augsburg, Nurnberg.	.....	45-70	Horizontal rotary gear	Hartung type oil relay, governor throttling steam; also automatic steam control on nozzle valves
British Westinghouse Co.	.....	.....	Rotary gear pump	Heavy Hartung type governor direct-connected to throttling valve
Bergmann-Elektricitäts-Werke.	55	110	Rotary gear pump	Hartung type governor with automatic steam control on nozzle valves
Belliss & Morcom.	40-50	As high as 90	Double-acting valveless pump, 80-100 r.p.m.	Centrifugal governor, oil relay simply throttling steam supply
Escher Wyss & Co.	.....	.....	Gear pumps, 1 for relay, 1 for bearing oils	Hartung type, oil relay simple throttling governor
Maschinenfabrik Oerlikon.	Not exceeding 66	Not exceeding 75	Rotary gear pump	Hartung type, oil relay governor, throttling steam supply; main throttle valve also opened by oil relay
Sächsische Maschinenfabrik.	.....	.....	Rotary type	Oil relay governor
Frazer & Chalmers.	.....	.....	.....	.....
Pokorny & Wittekind.	50-80	30-50	.....	Oil relay governor
General Electric Co.	.....	.....	Rotary gear pumps	Vertical centrifugal governor operating oil relay which controls nozzles; nozzle governing
British Thomson Houston Co.	50-60	50-70	Rotary pump	Same as General Electric Co

TABLE 1—*Continued*

Manufacturer	Oil Data	Type of Glands	Provision for Overspeed Regulation
C. A. Parsons & Co.	.....	Steam-packed labyrinth at shaft	.....
Westinghouse Mach. Co.	Oil pressure on bearings 3 to 4 lb., relay pressure 60 lb.	Water-packed glands with centrifugal	Emergency governor to shut off steam at about 12% overspeed
Willans & Robinson.	Bearing pressure about 10 lb.	Steam-packed labyrinth	Emergency governor for overspeed
Brown Boveri & Co., Richardsons Westgarth & Co.	.....	Steam-packed labyrinth	Emergency governor closes spring controlled throttle valve
Brush Electrical Engrg. Co.	.....	Steam-packed labyrinth; water-packed glands also used	Emergency oil relay control on throttle valve
Erste Brünner M. F. G.	7-10 lb. on bearing	Steam-packed labyrinth	Emergency governor to close spring loaded throttle valve
Franco Tosi.	10.4 gal. per min. pumped on 300 kw.; 18.4 gal. per min. on 12,000 kw.	Steam-packed labyrinth; water-packed glands also used	Emergency oil relay governor to close main throttle valve at 15% overspeed; unique device
Gebrüder Sulzer.	Bearing oil pressure 20 lb.	Steam-packed labyrinth; labyrinth packing consists of large number thin sheets placed radial, with close clearance on shaft	Oil relay control on steam inlet valve
Melms & Pfenninger, Breitfeld, Danek & Co.	Oil temperature at bearing 105 to 120°	Steam-packed labyrinth; exhaust end packing adjustable.	Emergency valve to operate at from 10-15 per cent overspeed
Allis-Chalmers Co.	Bearing pressure 5 lb., relay pressure 30 lb., bearing oil temperature 110° F. best; large quantities pumped	Water-packed glands with centrifugal impeller	Emergency governor to close spring-loaded throttle valve

Allgemeine Elektricitäts Gesellschaft.	Flood bearings	Steam-packed labyrinth	Emergency governor to close spring-loaded throttle valve
Maschinenfabrik Augsburg, Nurnberg.	.....	Graphite rings in glands, steam-packed	Emergency governor
British Westinghouse Co.	.....	High-pressure end, labyrinth and water gland; low-pressure end, water gland with centrifugal impeller	Emergency governor closes spring actuated safety valve on steam inlet
Bergmann-Elektricitäts-Werke.	Relay pressure 30 lb.	Steam-packed labyrinth	Emergency governor to close throttle at 10% overspeed
Belliss & Morcom.	Relay pressure 40 lb.; estimate 10 per cent of heat losses to go through oil, design oil pump accordingly	Multi-cellular with carbon rings and steam packed	Emergency governor controls oil relay to shut off steam
Escher Wyss & Co.	.....	High-pressure and low-pressure end, carbon ring packing	Emergency governor to operate at 8-10% overspeed
Maschinenfabrik Oerlikon.	Oil usually required is 0.02 gal. per kw-min.	High-pressure, graphite; low-pressure, steam-packed labyrinth	Emergency governor closes main throttle valve through oil relay
Sächsische Maschinenfabrik.	Oil pressure 2-2.5 atm.	Stuffing box, carbon rings	Emergency governor to operate at 15% overspeed
Frazer & Chalmers.	.....	High-pressure, metal rings and labyrinth; low-pressure, labyrinth	.....
Pokorny & Wittekind.	From 8-18 gal. per min. circulated according to size	High-pressure, carbon rings and labyrinth; low-pressure, labyrinth	.....
General Electric Co.	Relay pressure about 70 lb., bearing pressure 25 lb.	Carbon ring packing on both ends	Emergency governor to close unbalanced throttle valve
British Thomson Houston Co.	½ gal per min. per bearing on 1000 kw. units; 1½ gal. on 5000 kw. unit; same amounts to thrust	Carbon ring packing	Emergency governor to close throttle valve at 15% overspeed

TABLE 1—*Continued*

	Type of Coupling	Additional Notes
C. A. Parsons & Co.	Flexible claw	Simple Parsons turbines, usually very long spindles; large sizes in 2 cylinders
Westinghouse Mach. Co. Manufacturer	Flexible claw	Small sizes simple Parsons; large sizes Curtis-Parsons, double flow in low-pressure, heavy thrust block to take any end thrust
Willans & Robinson.	Flexible claw	.....
Brown Boveri & Co., Richardsons Westgarth & Co.	Flexible claw	Effect on steam consumption of changes in (*a*) vacuum, 5-6% per 1 in. vacuum; (*b*) steam pressure, 1% per 15 lb.; (*c*) superheat, 1% per 11° F.
Brush Electrical Engrg. Co.	Flexible claw	.....
Erste Brünner M. F. G.	Flexible claw	
Franco Tosi.	Flexible claw	Effect on steam consumption of changes in (*a*) vacuum, 7% per 1 in. at 28 in.; (*b*) steam pressure, 1% per 15 lb.; (*c*) superheat, 1% per 12.5° F.; large thrust block in form revolving piston, labyrinth packings; oil admitted both sides, movement shaft opens one side, closes other, making pressures on both different, automatically compensating for end thrust
Gebrüder Sulzer.	Special design of flexible coupling	End thrust all taken up by automatic oil controlled balance piston on end of shaft similar to Franco Tosi's
Melms & Pfennin-Danek & Co.	Flexible claw	Effect on steam consumption of changes in (*a*) vacuum 28-29 in., 5%; 27-28 in., 4%; 26-27 in., 3.5%; (*b*) steam pressure 1% per 12 lb.; (*c*) superheat 1% per 10° F.
ger, Breitfeld,		
Allis-Chalmers Co.	Flexible claw	.....

Allgemeine Elektricitäts Gesellschaft.	Solid	No vertical types
Maschinenfabrik Augsburg, Nurnberg.	Solid	.....
British Westinghouse Co.	Flexible claw	.....
Bergmann-Elektricitäts-Werke.	Solid and flexible claw	.....
Belliss & Morcom.	Special design of flexible coupling	Flood bearings with great quantity of oil to flush away heat rapidly
Escher Wyss & Co.	Solid	.....
Maschinenfabrik Oerlikon.	Flexible with leather disks	Blade angles, 20 deg. on stationary blades, 30 deg. on moving blades
Sächsische Maschinenfabrik.	Solid or flexible	.....
Frazer & Chalmers.	.....	.....
Pokorny & Wittekind.	Solid	Practically all turbines built for connection to turbo-compressors; effect on steam consumption of changes in (*a*) vacuum live-steam turbines, 5% per in., exhaust steam turbines 10% per in. vacuum; (*b*) steam pressure, 1% per 25 lb.; (*c*) superheat 1% per 12° F.
General Electric Co.	Solid	.....
British Thomson Houston Co.	Solid	Effect on steam consumption of changes in (*a*) vacuum 7% increase per in. above 28 in., 150 lb. steam gage, 8% decrease per in. below 28 in., 150 lb. steam gage; (*b*) steam pressure 1% per 10 lb. around 150 lb. pressure: (*c*) superheat 1% per 10° F.

TABLE 2 ECONOMY TESTS OF HIGH PRESSURE STEAM TURBINES

EFFICIENCY RATIOS BASED ON EFFECTIVE HORSEPOWER MARKS & DAVIS STEAM TABLES USED

Maker of Turbine	Type	Date of Test	Load-Kw.	R.p.m.	Steam Pressure Lb. Absolute	Temperature at Throttle, deg. fahr.	Vacuum referred to 29.92" Bar	Condenser Pressure, Lb. Absolute	Lb. of Steam per Kw-Hr.	B.t.u. per Kw-Hr.	Heat Utilized per Lb. of Steam	Heat Available per Lb. of Steam	Efficiency Ratio	Reference
Erste Brünner M. F. G....	Curtis-Parsons	1910	2128	1500	156.2	482	27.89	0.995	13.82	16460	247.0	343.8	71.8	Periodische Mitteilungen
Erste Brünner M. F. G....	Curtis-Parsons		6000	960	184.9	573	28.18	0.854	12.56	15570	271.5	380.7	71.3	Zeit. D.V.D. Ing., 12/10/'10
Erste Brünner M. F. G....	Curtis-Parsons	1910	7442	960	192.0	584	28.18	0.853	12.625	15705	270.2	384.4	70.3	Periodische Mitteilungen
Westinghouse Machine Co.	Curtis-Parsons	1910	9173	1800	181.7	433	27.81	1.032	14.57	16925	234.1	340.2	68.9	Trans. A.S.M.E., vol. 32
Brown Boveri & Cie.....	Curtis-Parsons		3053	1360	150.2	505	29.00	0.456	13.01	15990	262.2	385.5	68.0	Dinglers P.J., 6/17/'11
Erste Brünner M. F. G....	Curtis-Parsons	1910	1416	1260	128.2	482	27.60	1.137	15.18	18060	224.6	326.5	68.8	Periodische Mitteilungen
Brown Boveri & Cie.....	Curtis-Parsons	1911	1750	1500	176.4	586	27.08	1.392	14.23	17500	239.5	354.8	67.5	Zeit. F.D.G. Turb., 5/30/'11
Brown Boveri & Cie.....	Curtis-Parsons	1910	3764	1500	161.2	561	28.77	0.562	13.04	16290	261.5	391.4	66.8	Zeit. F.D.G. Turb., 5/30/'11
Westinghouse Machine Co.	Curtis-Parsons		9830	750	192.2	475	27.22	1.322	15.15	17790	225.2	336.0	67.0	Trans. A.S.M.E., vol. 32
Brown Boveri & Cie.....	Curtis-Parsons	1911	1495	3000	200.6	563	26.41	1.720	14.78	17880	230.7	345.5	66.8	Data from Manufacturer
Brown Boveri & Cie.....	Curtis-Parsons	1911	1271	3000	172.1	568	27.31	1.278	14.61	17880	233.5	354.3	65.9	Data from Manufacturer
Westinghouse Machine Co.	Curtis-Parsons		11466	750	191.7	484	28.07	0.910	14.45	17210	236.0	360.5	65.5	Trans. A.S.M.E., vol. 32
Erste Brünner M. F. G....	Curtis-Parsons		1250	3000	184.9	573	27.89	0.996	14.32	17680	238.2	373.1	63.9	Zeit. D.V.D. Ing., 12/10/'10
Brown Boveri & Cie.....	Curtis-Parsons	1910	3320	1500	180.9	525	29.02	0.440	13.50	16680	252.7	401.3	63.0	Zeit. F.D.G. Turb., 5/30/'11
Brown Boveri & Cie.....	Curtis-Parsons		5128	1000	171.2	565	28.52	0.726	14.35	17830	237.7	382.9	62.1	Stodola, 4th ed., p. 449
Breitfield, Danek & Co.....	Impulse-Parsons	1909	3585	896	160.7	457	28.32	0.782	16.08	19070	212.0	352.4	60.2	Zeit. D.V.D. Ing., 12/10/'10
Brown, Boveri & Cie.....	Parsons	1910	6257	1210	203.7	559	29.02	0.440	11.95	14980	285.5	415.0	68.8	Official Test Report
Allis-Chalmers.....	Parsons	1908	4300	1800	186.4	484	27.96	0.960	14.02	16690	243.4	355.7	68.4	Sibley Jour. of Eng., 1/11'
Brown Boveri & Cie.....	Parsons	1903	3500	1360	156.4	499	28.84	0.532	13.71	16720	248.5	378.6	65.6	Zeit. D.V.D. Ing., 12/10/'10
Brown Boveri & Cie.....	Parsons		3000	1360	165.0	625	27.02	1.420	14.75	18433	231.3	359.5	64.3	Die Turbine, 6/20/'11
C. A. Parsons & Co.....	Parsons		5164	1200	214.3	509	28.95	0.473	13.18	16140	258.7	402.3	64.3	Stodola, 4th ed., p. 439
Allis-Chalmers.....	Parsons	1911	3850	1800	164.7	491	27.91	0.983	15.40	18410	221.3	348.3	63.5	Power, 1/2/'12
A. E. G.....	Curtis-Rateau	1911	6518	1220	198.7	601	29.28	0.352	11.43	14640	298.4	434.2	68.7	Official Test Report
A. E. G.....	Curtis-Rateau	1911	6565	1220	200.2	597	29.18	0.406	11.64	14848	293.0	427.7	68.5	Official Test Report

British Westinghouse	Curtis-Rateau	1911	5066	1500	190.2	552	28.68	0.649	13.00	16100	262.4	391.5	67.0	Electrical Review, 6/23/'11
M. A. N.	Curtis-Zoelly		3584	1500	178.3	569	27.54	1.166	13.99	17190	243.7	361.3	67.5	Data from Manufacturer
Bergmann	Curtis-Rateau	1909	1545	1500	188.5	581	28.59	0.654	12.97	16230	263.0	396.3	66.4	Zeit. D.V.D. Ing., 12/10/'10
Bergmann	Curtis-Rateau	1910	2477	1500	140.0	522	28.81	0.588	13.93	17135	244.8	373.4	65.6	Elec. Zeit., 4/20/'11
A. E. G.	Curtis-Rateau	1908	4239	1500	188.3	662	29.11	0.397	11.97	15620	284.9	439.0	64.9	Stodola, 4th ed., p. 404
British Westinghouse	Curtis-Rateau	1911	2930	1500	210.2	568	28.18	0.894	13.72	16935	248.7	383.3	64.9	Electrical Review, 4/28/'11
A. E. G.	Curtis-Rateau	1907	3169	1500	184.7	592	29.11	0.397	12.74	16230	267.7	425.1	63.0	Trans. A.S.M.E., vol. 32
M. A. N.	Curtis-Zoelly		2507	1500	175.5	460	27.40	1.234	16.24	19020	210.0	334.6	62.8	Data from Manufacturer
Bergmann	Curtis-Rateau	1911	3365	1500	171.0	536	26.00	1.98	15.09	17970	234.1	381.3	68.5	Official Test Report
James Howden & Son	Zoelly	1909	6383	1000	202.7	520	27.33	1.269	14.305	17150	238.5	353.0	67.5	Engineer, London, 10/29/'09
M. A. N.	Zoelly	1910	1400	3000	180.7	554	27.40	1.237	14.21	17310	240.0	356.2	67.4	Zeit. D.V.D. Ing., 12/10/'10
Escher Wyss & Co.	Zoelly	1910	2052	3000	193.9	585	28.39	0.750	13.04	16290	261.5	392.6	66.6	Zeit. F.D.G. Turb., 2/20/'11
Escher Wyss & Co.	Zoelly	1910	4189	1000	179.7	557	28.66	0.618	13.30	16520	256.5	391.3	65.5	Zeit. F.D.G. Turb., 2/20/'11
F. Ringhoffer	Zoelly	1908	3000	1000	170.7	470	27.60	1.138	15.52	18278	219.8	339.2	64.8	Zeit. D.V.D. Ing., 12/10/'10
M. A. N.	Zoelly	1910	1250	3000	182.1	582	28.82	0.540	13.09	16500	260.2	404.5	64.4	Zeit. D.V.D. Ing., 12/10/'10
Oerlikon	Rateau	1911	3166	1500	213.9	663	29.25	0.367	11.44	14970	298.2	450.6	66.1	Engineering, 10/20/'10
Escher Wyss & Co.	Zoelly		5118	1000	133.7	549	27.55	1.161	15.18	18530	224.6	341.6	65.7	Dinglers P.J., 7/15/'11
Escher Wyss & Co.	Zoelly	1908	5000	1000	166.4	539	26.38	1.736	16.13	19350	211.2	330.4	63.9	Zeit. D.V.D. Ing., 12/10/'10
Escher Wyss & Co.	Zoelly		3540	1500	155.1	469	28.21	0.838	15.07	17940	226.3	349.5	64.8	Dinglers P.J., 7/15/'11
Escher Wyss & Co.	Zoelly	1910	1641	3000	221.0	672	27.91	0.985	13.08	16775	260.6	406.5	64.1	Zeit. F.D.G. Turb., 2/20/'11
Escher Wyss & Co.	Zoelly	1910	1235	3000	176.8	451	28.39	0.750	15.35	18156	222.3	357.8	62.2	Zeit. F.D.G. Turb., 2/20/'11
British Thomson-Houston	Curtis	1911	2987	1500	154.7	505	26.75	1.557	15.96	18960	213.7	321.2	66.5	Engineering, 10/20/'11
Gen. Elec. Co.	Curtis		3464		210.0	513	28.75	0.575	13.62	16620	250.4	393.4	63.6	Trans. A.S.M.E., vol. 32
British Thomson-Houston	Curtis	1909	2500	1500	126.5	414	28.47	0.711	15.92	18590	214.0	336.1	63.7	Zeit. D.V.D. Ing., 12/10/'10
A. E. G.	Curtis	1906	3000	1500	191.3	590	29.05	0.427	12.79	16240	266.6	420.4	63.4	Zeit. D.V.D. Ing., 12/10/'10
A. E. G.	Curtis	1909	2236	1500	191.6	654	29.34	0.284	11.77	15450	289.8	455.8	63.6	Zeit. D.V.D. Ing., 12/10/'10
Gen. Elec. Co.	Curtis		8880		192.5	487	28.02	0.933	15.05	17965	226.7	359.5	63.1	Trans. A.S.M.E., vol. 32
British Thomson-Houston	Curtis	1911	1541	1500	149.7	365	27.97	0.956	17.46	19720	195.3	320.2	61.0	Engineering, 10/20/'11
Gen. Elec. Co.	Curtis		10816	750	190.0	525	29.39	0.260	12.90	16135	264.5	427.3	61.9	Trans. A.S.M.E., vol. 32
Gen. Elec. Co.	Curtis		5095		185.1	554	29.40	0.255	12.71	16090	268.4	436.0	61.6	Trans. A.S.M.E., vol. 32
British Thomson-Houston	Curtis	1911	1221	3000	134.7	448	27.16	1.353	17.75	20690	192.2	314.0	61.2	Engineering, 10/20/'11
Gen. Elec. Co.	Curtis	1910	8775	750	194.0	451	27.95	0.956	15.95	18720	213.8	350.8	61.0	Trans. A.S.M.E., vol. 32

References: Zeit. D.V.D. Ing.—Zeitschrift des Vereines Deutscher Ingenieure; Zeit. F.D.G. Turb.—Zeitschrift für das Gesammte Turbinenwesen; Dinglers P.J.—Dinglers Polytechnisches Journal; Elec. Zeit.—Electrotechnische Zeitschrift.

72 The superiority of the Curtis-Parsons over the Parsons type is probably due to the reduction in the fluid friction and rotational losses occurring in the first cylinder of the Parsons by the use of a Curtis stage in this section.

73 The Parsons low-pressure sections evidently utilize the heat in the steam only slightly more efficiently than do the impulse turbines. The great surface areas of all disk type turbines which must be whirled in steam, produce losses which are apparently somewhat larger than the combined whirling losses and leakage in the Parsons drum turbines. Both the Zoelly and Curtis-Rateau types appear to use the steam more effectively in the low-pressure sections than the Curtis alone. Many European engineers hold the opinion that where high economy is to be obtained, the impulse turbine of the Rateau or Zoelly type is superior to the Curtis, though its manufacturing costs are higher. The Curtis-Rateau construction has all the commendable features of impulse turbines and has proved very economical.

74 The results shown in Table 2 are from the best reliable tests that have been made on each type. Objection may be raised that these results do not represent actual operating conditions as under varying loads, nor the average economy of any type of turbine. For instance, the Curtis turbine usually gives a very flat water-rate curve, while the Parsons type is more convex. On the other hand, recent tests on the new Curtis-Parsons types have also shown flat water-rate curves at various loads. It was impossible to compare the various types from this standpoint on account of absence of complete data of such tests. Melms and Pfenninger state in a recent communication that the turbine of their type built by Breitfeld, Danek & Company, and reported in Table 2 to have an efficiency ratio of 60.2 per cent was built for a special purpose and operated below its most efficient speed. They do not consider it representative of the best performance of their type and submit the following values of efficiency ratios which they will now guarantee on their turbines of the Curtis-Parsons type: For 1000 to 3000 kw. turbines r.p.m. 69 to 73 per cent efficiency ratio; for 1500 to 5000 kw. turbines at 1500 r.p.m. 68 to 72 per cent efficiency ratio. These figures would indicate a careful study of their machines and decided improvements in recent designs.

75 It is interesting to note in Table 2 that the best results have

been obtained within the past two years and that these show a considerable increase in efficiency over the earlier turbines.

LOW AND MIXED-PRESSURE TURBINES

76 One of the first low-pressure turbines installed was described by Professor Rateau in his paper on Different Applications of Steam Turbines.[1] The exhaust steam from various non-condensing reciprocating engines around mines was conducted to a regenerator, from which the turbine drew its steam supply.

77 Many low-pressure turbines have been erected since 1904 and have shown very economical results. In some installations it is usual to provide for operation on high-pressure steam when the supply of exhaust steam is insufficient to meet the power demand. Hence, the mixed pressure turbine has been developed. In Europe a Curtis stage is added at the inlet and the live steam passed through this before entering the low-pressure section. The whole of the heat content of the live steam can be effectively utilized by this method. Bleeder turbines are also being built, in which, after partial expansion to some fixed pressure, a portion of the steam is withdrawn from the casing for heating or industrial purposes.

78 Low-pressure turbines are frequently installed to use the exhaust steam of reciprocating engines without regenerators. In this case the generators are sometimes tied together electrically and the turbines are only fitted with an overspeed governor. Messrs. Stott and Pigott showed the results that could be obtained from such a combination in their paper, Test of a 15,000-Kw. Steam-Engine-Turbine Unit.[2] Such turbines are usually installed in stationary work, only when the reciprocating engines are already in service. The high-pressure turbine in a new plant requires less floor space, has less complicated machinery, is cheaper in first cost and in maintenance, and approaches, if it does not equal, the economy to be derived from the combination unit in every-day service. Nevertheless, there have been a number of combined engine and turbine plants recently installed in England which have proved very satisfactory.

79 It is probable that low-pressure turbines will be installed in the near future in large gas engine stations to utilize the waste heat in the gas engine exhausts.

[1]Trans. Am. Soc. M. E., vol. 25, p. 782.
[2]Trans. Am. Soc. M. E., vol. 32, p. 69.

TURBO-COMPRESSORS

80 Turbo-compressors have some decided advantages over reciprocating compressors, such as smaller floor space, absence of inlet and discharge valves, low cost of upkeep and no internal lubrication. They are being manufactured quite extensively in Europe and have been introduced in America by the General Electric Company, one of whose turbines was described by Mr. R. H. Rice in his paper, Commercial Application of the Turbine Turbo-Compressor,[1] read at the Pittsburgh meeting of the Society.

81 Turbo-compressors are built either with curved or radial impeller blades which discharge the air into smooth expanding diffusor channels to convert the velocity energy of the current into pressure. Usually guides are provided to direct the air into the entrance of the next stage without eddies. All passages are made as smooth as possible with no abrupt bends or turns, and all walls are water-cooled. The air is prevented from leaking back from stage to stage by labyrinth packings such as are used in impulse turbines.

82 European builders have installed turbines to deliver air at as high pressures as 130 lb. per sq. in. gage and are prepared to furnish them up to 180 lb. discharge pressure.

83 Turbo-blowers have also been built to deliver large volumes of air at low pressure, such as are required in furnace work. These units have no water-cooled jackets. The first difficult problem encountered in the construction of turbo-blowers or compressors was the provision of a suitable governing device for the unit. However, several ingenious and satisfactory arrangements have recently been developed and it is probable that this difficulty will soon be overcome completely.

84 The efficiency of turbo-compressors with water-cooling is defined as the ratio of the power required to compress the given quantity of gas isothermally to the power consumed at the compressor coupling in the actual compression. This efficiency in well designed units with discharge pressures between 60 and 150 lb. should fall within the limits of 60 and 70 per cent. The best results noted up to the present time were obtained on a turbo-compressor built by Pokorny and Wittekind for the Victoria Falls Power Company in South Africa, which on official test showed an efficiency of 67.9 per cent and on another unit furnished to the same company by the Allgemeine Elektricitäts Gesellschaft which gave an efficiency of 67.9 per cent when working against its normal discharge pressure.

[1] The Journal Am. Soc. M. E., March 1911, p. 301.

The A. E. G. have now under construction three turbo-compressors for South Africa, each of which will deliver 49,400 cu. ft. per minute at a pressure of 127.8 lb. per sq. in. These machines, each of which will be directly connected to a 10,000-h.p. steam turbine, will have a capacity at least twice as great as any turbo-compressor built hitherto.

85 When there is no water-cooling provided, the efficiency of a turbo-blower is expressed as the ratio of the power required to compress the given quantity of air adiabatically to the power actually expended at the compressor coupling. This efficiency, depending on the size of the blower, should fall between 70 and 80 per cent as a maximum. An efficiency of 78 per cent has been obtained on official tests of a Rateau turbo-blower built by Kuhnle, Kopp and Kausch and is probably the best result obtained up to the present time on this type of compressor.

86 It can thus be seen that in so far as efficiency is concerned, the turbo-compressor is equal to the average reciprocating compressor. It seems probable that turbo-compressors and blowers will be used to an increasing extent, largely on account of their low first cost and operating costs as compared with steam reciprocating units.

87 The high thermal efficiency of the gas-driven blowing engine exceeds that possible in a turbo-compressor unit, so that the former will continue to be used in blast-furnace work.

TURBO-DRIVEN PUMPS

88 The steam turbine is an ideal source of power to drive centrifugal pumps, especially when it is necessary to lift against high heads. Hence it has been installed in several places for city fire service, using lake or river water in high-pressure mains. The efficiency of such centrifugal pumps usually ranges from 65 to 80 per cent, so that, in spite of the high efficiency of the turbine itself, the combined set will not give as good economy as a high grade reciprocating pumping engine. However, its first cost is low, it requires no internal lubrication, takes up very little floor space, and has no valves to require examination or renewals.

GEARED TURBINES

89 Attempts have been made to adopt steam turbines for direct connection to continuous current generators and other slow-speed machinery. A steam turbine to be economical must be a high-speed machine, and hence its use with slow-speed machinery has not proved entirely satisfactory. Dr. De Laval adopted spur gearing as

a means of reducing speeds on his first simple impulse turbines, and builders of this type still use this construction. It is only within the last few years that attempts have been made to apply gearing as a means of speed reduction on other types of turbines. The Westinghouse Machine Company are now manufacturing direct-current turbo-generators with the Melville-MacAlpine reduction gearing between turbine and generator. This gear is also being built for use in marine work to drive slow-speed propellers. The gear wheels in this construction are carried on a floating frame so that the teeth may be always in correct alignment. One set of such gearing showed under test an efficiency of 98.5 per cent.

90 C. A. Parsons & Company have built several notable reduction gears for steam turbines, in which the floating frame idea was omitted. A mixed pressure steam turbine of 750 b.h.p. is now in use driving a three high set of rolls through gearing at the Calderbank Steel Works near Glasgow, Scotland. A flywheel is placed on the same shaft as the driven gear and thus takes the shock off the turbine when a billet enters the rolls. In 1909 the Parsons Marine Steam Turbine Company installed a geared turbine of 1000 h.p. in the S. S. Vespasian of 4000 tons. Extensive experiments were carried out and it was found that the efficiency of this gearing, which had no floating frame, ranged between 98 and 99 per cent. After a year's operation, in which the ship covered 20,000 miles, tests were again made on the gearing with equally good results. On examination no appreciable signs of wear could be noted on the gear teeth, which were made of mild chrome nickel steel and were flooded with oil.

91 It is perfectly feasible to adapt the steam turbine through gearing to belt and rope drives when these are required in large powers. Under certain conditions, where fuel costs are high and water for condensation is plentiful, such an installation would prove an economical investment in place of reciprocating engines. The direct turbine drive would then come into competition with the motor drive. The losses in lineshafting can be greatly reduced by the use of ball or roller bearings so that the turbine drive may prove very economical in some instances as compared with individual motor drives.

92 The use of geared turbines in large sizes has been in the nature of an experiment until quite recently. Judging from the results obtained in recent installations, their commercial success seems now assured. In the design of these units the tendency will

be to simplify details. The success of the Parsons gears on the S. S. Vespasian should encourage designers to do away with any special devices to secure alignment and to provide simply accurately cut gears properly meshed and running in a flood of oil.

MARINE TURBINES

93 All the standard types of turbines have now been adopted for marine service in driving screw propellers, either direct-connected or through gearing. The design of direct-connected turbines is complicated by the fact that the speed of screw propellers must necessarily be low as compared with the most favorable speeds for economical steam-turbine operation. Hence these turbines require large spindle diameters and massive construction and yield correspondingly poor steam economies, especially at slow speeds. These units are usually built with the power divided between two or more shafts connected to high and low-pressure cylinders. With geared equipments the turbines can be operated at their most efficient speeds, while the gears can be so designed that the propeller also runs at its most economical speed.

94 Reversing is made possible by suitable blading in the low-pressure ends of the main units into which live steam is admitted when desired. When the turbine is running forward, this blading revolves in vacuum and consumes but little power.

95 Combined types of turbines are also being introduced in marine installations. A recent Curtis design includes a drum impulse section. Mr. Zoelly now uses Curtis stages in his high-pressure section, but with steam velocities not exceeding 1300 ft. per second, obtained by converging nozzles only, and drum impulse construction on the low-pressure ends. Several other European builders have also used Curtis stages in the high-pressure portion. Melms and Pfenninger have constructed marine turbines of 12,000 to 16,000 h.p. having impulse wheels with partial admission at the high-pressure end and Parson's blading in the remainder of the turbines.

96 Reciprocating engines exhausting into low-pressure turbines have been installed in several ships, the most notable of which is the S.S. Olympic, White Star Line, and have shown very satisfactory results. In this case the engines are used for reversing.

97 Many schemes have been proposed to install turbo-generators of central station type on shipboard and to operate the propeller shafts by means of large slow-speed induction motors. Marine engineers object to this arrangement on account of the dangers accom-

panying the use of such electrical machinery and auxiliaries in marine service. This objection seems to be due largely to inexperience with electrical machinery, as the essential conditions of operation in marine work do not differ greatly from those under which many electrical machines operate satisfactorily in land practice.

98 Turbo-driven lighting sets and other auxiliaries are being used in increasing numbers on shipboard, owing largely to the high efficiencies which may be obtained, to the small floor space required, and to the light weight of the units.

99 Recent orders for turbine-driven steamships abroad include some interesting equipments. The Canadian Pacific Railway has ordered two boats with four screws and with Parsons turbines. The two outside screws will have high and intermediate pressure turbines respectively, while the two center screws will be connected to low-pressure turbines. Two twin-screw passenger boats of 5000 h.p. with Parsons turbines have been ordered for the Southampton-Havre service. Each set consists of a high-pressure and a low-pressure turbine geared individually to its own propeller shaft. The British Government has ordered two twin-screw destroyers also to use Parsons geared turbines, totalling 14,000 h.p. per ship, or 7000 h.p. per gear. The United States Government has placed an order with the Westinghouse Machine Company for a gear equipment on one of its colliers.

100 Geared turbine equipments are thus making rapid headway on account of the high efficiency of the combination and the resultant favorable steam consumption obtained.

101 At the present time about 90 per cent of the marine turbines built have been of the Parsons type. Here again the inefficiency of the Parsons high-pressure sections has become apparent, so that it is probable that a construction similar to Zoelly's high-pressure end will be introduced in this section. Sir Charles A. Parsons is quoted as saying: "In the low-pressure blades of the Mauretania the leakage was practically nothing and their efficiency was about 85 per cent." [1] Under such conditions it does not seem probable that much higher efficiency can be obtained by use of other constructions than the Parsons reaction type in the low-pressure section. It is reported that the low-pressure blading of Parsons turbines in ships of the United States Navy has given considerable trouble, and also that

[1] Engineering, Oct. 27, 1911.

the turbines need more careful handling when starting up than do impulse turbines.

TREND OF TURBINE DEVELOPMENT

102 The cost of manufacture is a very important item in determining the future development of the steam turbine. Types such as the original Parsons and the Rateau, while inherently of very high efficiency, have too high manufacturing costs to compete with the newer combined types.

103 The writer offers as his opinion that the combined types, such as the Curtis-Parsons, the Curtis-Rateau and also the Curtis-Rateau-Parsons, previously described, will very soon supersede the simple types. It is probable that the Curtis turbine will eventually be built only in horizontal units and will gradually be modified to a Rateau or even a drum impulse construction in the low-pressure sections. The freedom from close adjustment in impulse turbines and the recent improvements in blading materials will greatly increase the use of this type, although Curtis-Parsons turbines are said to be cheaper to manufacture. In actual operation, it is an open question among engineers whether the reaction turbine has a higher commercial efficiency than the impulse type, and hence buyers usually consider first cost and personal preference only.

104 Turbines will probably be made shorter with very stiff shafts. With this construction many of the earlier blading troubles will disappear. But the peripheral speeds will also be increased and this will involve the development of suitable blading material and methods of holding blades that will satisfy these new requirements. Recent results seem to indicate that improved efficiency may be looked for with increased blade speeds.

105 Several impulse turbines have been built recently in Europe where the expansion was not complete in the nozzle, so that a portion of the expansion took place in the first moving blades. Some large Curtis turbines recently installed in America are said to have Parsons blading in the last stage. These developments would indicate a movement to introduce reaction principles in impulse turbines, and further illustrate the tendency to merge types.

106 The hope of further improvement in efficiency lies in extensive study, particularly of the action of the steam during its passage through the moving and stationary blades, of the effect of form of blades, passages and casings and of various forms of baffles and balance pistons to prevent leakage. Such research work has not

been carried out up to the present time by most manufacturers, largely on account of the extreme care and heavy expense involved in such tests. The present state of development has been largely one of cut and try. The increasing competition of the gas engine and the possible development of a satisfactory gas turbine will force manufacturers to develop their turbines to the greatest degree of economy.

107 With regard to detail, simplicity will be the leading consideration. With the introduction of Curtis high-pressure stages, nozzle governing will undoubtedly be used to an increasing extent, though the results obtained by Westinghouse, by Zoelly, and by Bergmann with simple throttling governors, raise a question as to whether the additional complication of nozzle governing will pay. Oil relays will probably replace all other systems of governing on account of their simplicity and reliability. The simple and efficient centrifugal oil pump governor of Sulzer appears to be an improvement of considerable moment, and will probably receive extensive use.

108 With the development of suitable gearing for steam turbines, their field of application has been greatly increased and turbines will shortly be used for purposes which engineers today would consider them utterly unfit. Low-pressure turbines will continue to be installed in plants where reciprocating engines are still in operation and also where large quantities of waste heat are available. The low and mixed pressure types of turbines will find a very extended use in connection with heating systems, evaporators, etc.

109 The development of the past ten years has been truly marvellous. No great gain in thermal efficiency seems possible, so that future improvements will be largely along the line of detail construction and modification.

110 The addresses of the firms referred to in Table 1 are as follows: C. A. Parsons & Co., Newcastle, England; Westinghouse Machine Co., East Pittsburgh, Pa.; Willans & Robinson, Rugby, England; Brown Boveri & Co., Baden, Switzerland; Richardsons Westgarth & Co., Hartlepool, England; Brush Electrical Engineering Co., Loughborough, England; Erste Brünner, Brünn, Austria; Franco Tosi, Legnano, Italy; Gebrüder Sulzer, Winterthur, Switzerland; Melms & Pfenninger, Munich, Germany; Breitfeld Danek & Co., Prague, Austria; Allis-Chalmers Co., Milwaukee, Wis.; Allgemeine Elektricitäts Gesellschaft, Berlin, Germany; Maschinenfabrik Augsburg Nürnburg, Nürnburg, Germany; British Westinghouse Co., Manchester, England; Bergmann-Elektricitäts-Werke, Berlin, Germany; Belliss & Morcom, Birmingham, England; Escher Wyss & Co., Zurich, Switzerland; Maschinenfabrik Oerlikon, Oerlikon, Switzerland; Sächsische Maschinenfabrik, Chemnitz, Germany; Frazer & Chalmers, Erith, England;

Pokorny & Wittekind, Frankfort-Bockenheim, Germany; General Electric Co., Schenectady, N. Y.; British Thomson Houston Co., Rugby, England.

The writer wishes to express his thanks to manufacturers of steam turbines in America and in Europe, who liberally provided him with the information on which this paper is based, both during personal visits to their works and by correspondence.

F. Hodgkinson. In the table showing the characteristics of turbines of different manufacturers there are certain points that the writer would like to add to or correct in reference to the machines built by the Westinghouse Machine Company, first respecting speeds.

Speeds have been increasing lately for 60-cycle work. For sizes of from 3000 to 10,000 kw. 1800 r.p.m. is the speed employed and for larger powers, 1200 r.p.m. For 25-cycle work, 1500 r.p.m. is employed up to a rating of 15,000 kw. (normal), or 20,000 kw. (maximum). A speed of 750 r.p.m. is not likely to be employed for capacities of less than 30,000 or 40,000 kw.

The speed of impulse blading, as employed by the Westinghouse Company, varies with the different types from 350 to 500 ft. per second. The speed of the reaction or Parsons blading at the low pressure depends upon the steam volumes involved and varies from 300 to 500 ft. per second, the latter in the case of large machines where constriction in the exhaust must be avoided on account of the large volume.

The journal speeds are higher than Professor Christie points out. Journal speeds reach 80 ft. per second and pressures 100 lb. per sq. in. of projected area, and this pressure may be materially increased without encountering wear of the bearings, provided the proper flooded lubrication is employed and bearings are sensibly designed. With higher surface speeds a little more oil-cooling capacity will be necessary on account of the greater rate of shear of the oil. A good warm temperature, not less than 150 deg., is conducive to high economy.

Concerning the economies of turbines, Professor Christie quotes the opinion of European engineers, that higher economy is to be obtained from impulse elements of the Rateau or Zoelly types, rather than with impulse elements of the Curtis design. The writer believes this to be a generally accepted fact and in the case of the latter type it is doubtless due to the low efficiency of the last row of moving blades. Apparently having once ex-

panded, steam eddies are brought about each successive time the steam is put through the blade elements. Somebody has remarked that the fourth row in a Curtis element has never been found useful. Theoretically it is useful, but in practice, except for very high pressure drops, it has not been found so.

It would seem that the reaction turbine, unless it be subjected to some disability, may be more economical than either of the other types. With any impulse turbine there are necessarily two complete conversions of energy, the first expanding the steam through the nozzle, the steam doing work in giving itself velocity; and the second extracting the velocity of the steam. Both of these transformations of energy are subject to losses, the second greater than the first. If the nozzles were permitted to revolve, and the turbine had no blades at all, obviously such a turbine would be the more efficient. The reaction or Parsons turbine is such a machine, but, unfortunately, it is subject to leakage. However, with higher speed machines and capacities that lend themselves to good proportions of blading, this leakage, particularly at the low-pressure end, is a very small factor, and such a turbine is the more efficient. These features are emphasized in the later high-speed turbines, particularly of the combination type, comprising a Curtis turbine element for the high pressure, and a reaction element for the low pressure, the Curtis element as referred to the regular Parsons, replacing a large number of rows of reaction blading at a small diameter with the attendant greater axial length, or a smaller number of rows of larger diameter, the blades being smaller in the latter case, and of poor proportions, subject to high leakage.

Of late, the tendency has been toward higher speeds, which, within reasonable limits, is conducive to better results, not only from the standpoint of economy, but also from the standpoint of operativeness. As Professor Christie has stated, congestion at the low-pressure end may be entirely avoided by making the last portion double-flow. Higher rotational speed involves smaller diameters, and hence better blade proportions at the high pressure end, with the attendant lesser leakage. In addition to this, the turbine cylinder structure as a whole is much smaller, there is less distortion of structure with temperature changes, thus permitting running the blades with smaller clearances if it is desired to do so.

Professor Christie very properly points out that there is a

lower leakage ratio in a single-flow turbine than in one of double-flow construction. However, the choice of revolutions is limited on account of the standard frequencies employed. While reducing the speed to the next lower one available may permit of single-flow construction, the advantages may be entirely offset by the increased clearances which must be employed, simply because the size of the machine has so materially increased, with the attendant greater amount of distortion. Therefore, the turbine having the low-pressure sections double-flow is frequently

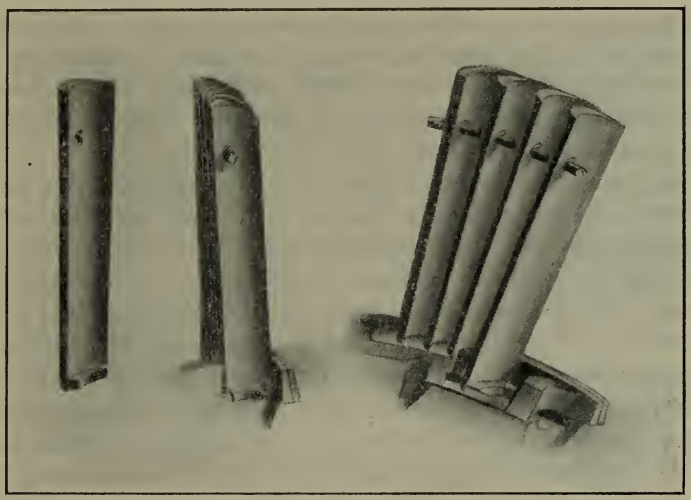


FIG. 13 · BLADING USED IN WESTINGHOUSE TURBINES

the more economical machine. It is very convenient to be able to double up the steam passage at the low-pressure end, because of the tremendous rate with which the volume increases, particularly with machines designed for high vacuum.

Concerning blade troubles and blade construction, the writer ventures the statement that no turbine built by a reputable manufacturer in this country ever came to grief because of the blades becoming detached or breaking, due in itself to centrifugal force. Accidents have occurred because of misalignment or distortion, resulting either in blades colliding with turbine parts, or with one another, which of necessity resulted in injury. The principal source of blade trouble is due to the blades buzzing or vibrating in the steam current. The natural vibrating period of the

blade may be raised by means of a shroud, or other form of lashing, but not eliminated.

Fig. 13 shows what has now become the standard construction for reaction blading, as employed by the Westinghouse Machine Company. The feature is an interlocking system, the strength of which is equal to that of the blade itself. The blade sections are stronger at the root than at the tip. The section tapers slightly from the root, thus eliminating the chance of injury due to any vibrating or buzzing, should it exist.

As shown in Fig. 13, the bottom end of the blade is upset, which is truly an upsetting process, and not any mere bending, and the die for holding the blade at this time is so arranged that the section of the blade is increased in thickness throughout its lower portion, this latter feature, however, not being clearly shown in the illustration.

The dove-tailed groove in the blade-carrying member is provided with a supplementary groove in the floor of the main groove, into which the upset end of the blade fits. The latter locks underneath the packing pieces which occupy the main groove. The larger blades are provided with compound wedges which go between the walls of the groove and the blades and packing pieces, which effectively fills the groove in a more satisfactory manner than a calking strip. In the case of the smaller blades, the wedges are found to be unnecessary, owing to the lighter sections of the packing pieces, which permits them to fill the groove, when driven into place. Inasmuch as no calking of any kind is resorted to, blades may be taken out and replaced an indefinite number of times.

The author has brought out an important point in the matter of standardization of ratings for turbines, and suggests that the Society fix a rating for turbines, defining their overload capacity. While this may be a little difficult to do, the writer cordially concurs with him that a great deal of confusion exists.

In discussing capacities of machines, one is compelled to go to some explanation as to what is meant. The kilowatt rating needs as much explanation as does the load factor.

Present methods cause particular confusion in the statements of the cost per kilowatt of the installation of a turbine station. To standardize ratings may be difficult unless the turbine be dissociated from the generator. There are, however, two needs of ratings for power-plant work today. One is for the very large

power plants, such as the Edison plants, which have a large number of units and large capacity, and the condition is such that if the machine is running at all it is running at its highest load. Obviously a generator in such a plant may advantageously be given a certain maximum continuous rating at a specified temperature rise, and the turbine connected to it arranged to give its highest economy at that load, and need have no overload capacity, or at most 10 or 15 per cent.

The other kind of rating required is that needed by the smaller plants with few units, the plant having to encounter heavy peaks at certain times of the day. Here, while the generator may have a certain maximum capacity as to its basis of rating, the turbine should have its highest economy at some fraction of this load. For such an application the old method of rating machines at a normal full load, with a 60-deg. rise, would seem to apply to the turbine having its best economy at that load and the generator capable of carrying 25 per cent overload continuously and both turbine and generator capable of running for a limited time at 50 per cent overload.

That system of rating was first practised in this country in the early days of turbine development and had not been practised in Europe. It was brought about by the turbine having to compete with the Corliss engines. Latterly, bargain-counter methods of buying (an expression borrowed from Professor Christie) have caused such outfits to have their turbines sometimes given a maximum rating corresponding to the 50 per cent overload just referred to. It would seem entirely feasible to have generators uniformly rated on maximum continuous load basis; the turbine, however, to have its best steam consumption at some fraction of this, depending on the requirements of the installation, and with overload capacity to suit the maximum possible load on the generator.

J. A. Moyer. Interest in Professor Christie's paper, in the writer's opinion, centers about his tabulation of recent tests of steam turbines. Such a tabulation is, however, scarcely complete if it does not include the very elaborately detailed and accurate test of a Sulzer turbine, of the combined impulse reaction type, reported by Professor Stodola.[1] At the load showing the best economy, 2058 kw., the steam consumption was 15.21 lb. per kw-

[1] Zeits. des Vereines deutscher Ingenieure; October 28, 1911, p. 1799.

hr.; speed was 1501 r.p.m.; absolute steam pressure, 178.1 lb. per sq. in.; superheat 182 deg. fahr.; vacuum referred to 29.92 in. barometer was 28.41 in. (0.740 lb. per sq. in. absolute pressure). Calculated from these data the B.t.u. per kw-hr. on the basis explained in Par. 69, are 18,800; the heat utilized per pound of steam 224.5 B.t.u.; the available heat per lb. of steam 380.1 B.t.u.; and the efficiency ratio or the efficiency of the Rankine cycle is 59.1 per cent. The generator efficiency was low for a machine of this size. This is certainly a reliable test of a presumably well-designed turbine of the Curtis-Parsons type, and the results are not particularly favorable when compared with the data ordinarily published by manufacturers or their agents. It should be clear, therefore, that the adoption of a design consisting both of impulse and reaction stages along with excellence of constructive details does not necessarily give higher efficiencies of the Rankine cycle than other types. The Sulzer turbine, however, was laboring with one disadvantage in that it had three rows of moving blades and two rows of stationary blades instead of the modern Curtis construction employing two rows of moving and one row of stationary blades per pressure stage. Furthermore, it has never been shown that anything is gained by attaching to a blade wheel more than two rows of moving blades or buckets.

The writer felt that tests reported as showing Rankine efficiencies greater than about 68 per cent should be accepted only with a great deal of caution. To be more definite, he was not sure that the data of the first, third and fifth tests reported in Table 2 might be accepted without reservation. All of these tests are nothing more than manufacturers' statements, not properly verified. At a glance one might suppose that the second of these tests, taken from a descriptive article in the Zeitschrift des Vereines deutscher Ingenieure, came with some basis for consideration, but the test itself does not bear the earmarks of one made with very much accuracy. Practically all the data are given in round numbers, usually indicating approximation. As stated, the load is exactly 6000 kw., the steam pressure is given roughly at 13 atmospheres absolute, the steam temperature averaged apparently exactly 300 deg. cent. at the throttle throughout the test, and the steam consumption per kw-hr. was calculated not much closer than the nearest half-pound. As regards the fourth test cited, the writer is inclined to agree with Mr. Emmet in his statement at a previous meeting of the Society when this test was being

discussed, that the "data are not altogether convincing." It seems to me that we should be very careful about incorporating into the Transactions reports of tests of boilers, engines and turbines supported only by the claims of manufacturers. It is very desirable that such tests should be made and reported by well-known engineers who have no financial interest in the business.

The statements in Par. 10 are not very clear as regards erosion or cutting of the blades. There is very little erosion of the blades in either the Curtis or the so-called Curtis-Parsons types if constructed of what are now considered by experts as suitable materials. The constant erosion of *continuous operation* for five years will have very little effect on the economy of the turbine as compared with the corresponding deterioration of a reciprocating engine.

The writer objected to another manufacturer's claim as given in Par. 51, where it is stated Mr. Zoelly "claims he gets the best results on his turbine by throttle governing." The Zoelly method of governing is cheap and simple, and that as the writer understands it, is the reason for its adoption. The writer would like to be shown, however, by authentic tests that it is even nearly as good as some other types.

C. V. Kerr. In Par. 10 there is a statement as to the wear of the blading in the lower stages being due to the wetness of the steam. In the double-cup form of turbine in which the writer has been interested, the wear of the blading was found to be in the upper stages. This may be due to the fact that a greater proportion of the available energy was allowed to produce velocity in the upper stages, with the idea that part of it would be transferred to the lower stages, and thus equalize the power. Or it may be that the erosive power of the steam is greater in these upper stages, due to its greater density. It is a question whether it is possible to equalize this wear throughout the stages by an exact division of the available energy.

In his connection with the small turbine business the speaker had to do with the driving of auxiliary generators, pumps and blowers. He took the stand that his company wanted to know the maximum power to be developed by a turbine, then they would on their test put into the turbine a little more than that maximum power to make safe. It is very important for that class of work and he thought it should be for generator work also. A gen-

erator is supposed to stand a certain normal overload for a certain period. What is the use of putting into a turbine more power than is required to move the generator at that power and thus cause a burning out or other difficulties? The small steam turbine builders dropped into the habit of selling their turbines on the basis of power developed. With a casing of a given size, however, the power may be doubled or quadrupled by adding nozzles. But the small steam turbine is in competition with the steam engine and the electric motor, and they are generally sold on the power basis. It is the practice now on the part of at least two of the small steam turbine builders, to establish a minimum price for the turbine based on manufacturing cost and a rating upward on the basis of additional power developed.

Regarding the manufacturers' tests, there are members of the Society engaged in the manufacturing business whose study of testing methods is as intense and of as high an order as any who are doing testing. Questions are being raised as to standard methods of testing which come from manufacturers' engineers engaged in testing. There is no monopoly of engineering honor in one particular section of engineers. Although a manufacturer's engineer, the writer stands in favor of accepting all engineers' tests on their full value, for it is a mark of engineering honor and a recognition of honor in others. He also believes that it is a mistake on the part of any engineer to put out data that are tainted in any way, since this will react on the offender sooner or later.

George A. Orrok referred to Professor Moyer's expressed doubt of the figures given in Table 2 of the paper for the Erste Brünner turbine. The results had seemed to him to be extremely good, and he had investigated them by writing to the engineer who made the tests, Professor K. Koerner, rector of the German Technical High School at Prague, who vouched for the data.

Clarence P. Crissey.[1] The accepted practice is to divide turbines into two fundamental types, namely, reaction turbines and impulse turbines. While these names may not be free from criticism, they are well understood and seem preferable to the names Parsons, Curtis and Rateau. While the latter terms give credit for inventions or variations in design, they are not based upon fundamental principles and, therefore, seem undesirable.

[1] General Motors Co., Detroit, Mich.

If it is desired to subdivide impulse turbines according to the manner of utilizing the steam, the following classes are suggested:

a Impulse turbines with only one velocity stage in each pressure stage

b Impulse turbines with two or more velocity stages in each pressure stage

c Impulse turbines in which all pressure stages do not have an equal number of velocity stages

Classification of impulse turbines might be carried further in order to illustrate a point of construction often overlooked: that is, the method of conducting the steam leaving the last buckets to the succeeding nozzles. Various constructions are:

d Impulse turbines in which the steam leaving the last moving buckets passes directly from the bucket into the succeeding nozzles without great loss of velocity

e Impulse turbines in which the steam leaving the last moving buckets passes into the casing and after losing practically all of its velocity enters the next set of nozzles

f Impulse turbines in which the steam leaving the last moving buckets is utilized in some pressure stages according to the method described in *e* and in other pressure stages according to the method outlined in *d*

It follows from this classification that Rateau turbines fall in class *a-e;* Zoelly turbines fall in class *a-e;* Curtis turbines of the General Electric Company fall in class *b-c;* Curtis turbines of A.E.G. (large size) fall in class *c-f*.

The Zoelly turbine is similar in principle to the Rateau and it seems preferable to class the Zoelly turbine as a Rateau type just as the Allis-Chalmers is called Parsons and not Fullager.

It will be noted that in the paper the single-row stages of the A.E.G. turbines are said to be of the Rateau construction. The Allgemeine Elektricitäts-Gesellschaft, however, state that these stages are constructed in accordance with the 1896 patent description of Curtis which represents the immediate forerunner of the Rateau and Zoelly constructions. The difference between the A.E.G. single-row stages and those of Rateau lies in the manner in which the steam leaving the moving buckets is utilized. This difference has been noted in the classification of turbine types given above, that is, the A.E.G. single bucket row

stages are of type *d*, in which the steam leaving the moving buckets passes directly into the succeeding nozzles without great loss in velocity while the Rateau fall under type *e*, in which the steam is brought to rest before entering the next set of nozzles. The large A.E.G. turbines, therefore, appear to be of the Curtis type throughout, although not altogether of the type popularly known as Curtis.

The moving buckets of reaction turbines have in practice been supported upon drums and those of impulse turbines usually upon disks. The manner of construction does not, however, change the fundamental type and, therefore, the author's designation, in Par. 24, of the Belliss & Morcom turbine as a Curtis-Parsons construction is confusing because this turbine operates entirely upon the impulse principle.

In reading Par. 5*b*, the impression might be gained that high steam velocities are inherent with impulse turbines having two or more velocity stages. Such is not necessarily the case. If low bucket velocities are employed the spouting velocity of the steam is also low. Impulse marine turbines illustrate this point.

It might be inferred from the last part of Par. 12 that constructions such as the A.E.G. sacrificed efficiency to first cost. According to Table 2 the A.E.G. design is more efficient than the Rateau.

In Par. 22 reference is made to Fig. 3 showing a turbine with six pressure stages. It is stated that the A.E.G. build similar turbines up to 1000 kw. capacity. While the A.E.G. machines below 1000 kw. have two velocity stages in a pressure stage the writer understands that they have only two pressure stages.

It would be interesting if Professor Christie could tell whether Mr. Zoelly advances any reason for his claim of greater efficiency with throttle governing.

In connection with Par. 35 it might be observed that theoretically correct action of the steam in any impulse turbine occurs only at one set of conditions. For any variation in conditions or quantity of steam flowing, a variation from the theoretically correct action must occur. The flat economy curves of impulse turbines prove that this is not important from the standpoint of efficiency, and practice shows that the end thrust is not appreciably increased by departure from the theoretical conditions.

The drum construction in impulse turbines was probably first used by the A.E.G. in the turbines for the steamship Kaiser,

which was tested in 1905. This firm's experiments led them to state that the drum construction was advantageous, for the low bucket speeds existing in direct-connected marine turbines when the steam had expanded to 50 times its initial volume.

Using the fundamental classification given in this discussion there is only one type of "modified" or "combined" turbine, that is, the impulse-reaction machine.

It is interesting to examine the evidence as to which of the fundamental types is at present gaining in popularity. There is conclusive evidence that a large number of manufacturers formerly producing reaction turbines have adopted impulse blading in the high-pressure sections of their machines but there is little to indicate that the manufacturers of impulse turbines generally contemplate introducing reaction blading.

Par. 105 states that "several impulse turbines have recently been built in Europe where the expansion was not complete in the nozzles, so that a portion of the expansion took place in the first moving blades." The only turbines of this character which the writer knows of were abandoned and if others are being built they probably are more or less of an experiment.

In Par. 70 the following statement is made: "It is therefore apparent that the efficiency ratio alone will express in the best manner the degree to which the designer has approached ideal results in his turbine." While the efficiency ratio comes nearer than any other single figure to expressing the economic results it must be used with caution because:

a The steam conditions must be taken into account. The same steam turbine operating with an equal amount of available energy will not have the same efficiency ratio if the energy is obtained in one case by a moderate vacuum and high superheat, and in another case by a high vacuum and low superheat.

A non-condensing turbine may have the same or a better efficiency ratio than a condensing machine, but this is no indication that the non-condensing turbine is as suitable or better for a given plant than the condensing unit.

b It is more difficult to obtain a high efficiency ratio with a large amount of available energy than with a smaller amount. Referring to Table 2 it is seen that the 6000-kw. machine, built by Erste Brünner M.F.G.,

has an efficiency ratio of 71.3 and the heat available per pound of steam is 380.7 B.t.u. The A.E.G. turbine of about the same rating gives an efficiency ratio of 68.7, but the heat units available per pound of steam are 434.2 B.t.u. or about 14 per cent greater. If the latter machine were operated with steam having 380.7 B.t.u. per lb., there is little doubt but that the efficiency ratio would equal or approach closely that of the Erste Brünner turbine.

c *The efficiency of the electrical generators is included in the efficiency ratios.* This has considerable importance if the ratios are used to compare turbine types. It is stated in Par. 59 that European builders generally guarantee better generator efficiencies than American builders.

d The results may be affected by the inclusion or exclusion of the power necessary for field excitation and auxiliaries, as well as other variations in testing practice.

From a technical viewpoint the revolutions per minute of the turbines cannot be neglected in making comparisons upon which to base opinions as to types. Either type is benefited by an increase of rotative speed and it is almost certain that in the future the speed of all types will not differ materially for machines of a given rating and service.

Too much reliance must not be placed upon efficiency ratios, either in the selection of a turbine at the present day or as a basis of judgment regarding ultimate types because first cost, reliability, sustained economy, life and operating expenses are such important factors. The author points out, and it is well to emphasize the fact, that his figures for efficiency ratio do not represent operating conditions such as occur with varying loads or the average economy of any type of turbine.

E. D. Dreyfus emphasized the necessity of establishing some method of ratings which would be recognized by engineers at large, and also rules for conducting tests of this type of power generating unit, that is, including the turbine and generator as a whole. As the paper states the case there is now an undesirable dissimilarity in practice in rating this class of machinery.

The Society should lay down some guide or formulate some code which would create more uniformity and a better under-

standing in this connection. Failure to do this will leave room for criticism, which seems already to have shaped itself in the minds of some. It is well to note how this situation has been viewed by others, and the writer quotes a very brief abstract from an editorial from the Electrical Review and Western Electrician [1] under the heading of Ratings, which followed a paper he read one year ago on Turbines in which this vital question was seriously referred to:

There are a variety of bases upon which ratings may be made. The ultimate capacity of the machine is one basis, the load at which maximum efficiency is secured is another, the load at which deterioration is less is another, the load which is fixed by some limitation of operation is another, and so on. Most electrical apparatus is limited in capacity by the allowable temperature rise and the rating is made upon this basis. The Standardization Rules of the American Institute of Electrical Engineers fix the allowable temperature, and performance specifications are drawn to coincide with these rules, or else specify limiting temperatures considered more suitable for the given working conditions. * * *

Turning to steam engines we find the rating to be based upon operation at maximum efficiency and normal speed. * * * Practice is not entirely uniform here, however, and there is a certain leeway dependent upon personal opinion or the characteristics of the particular engine. In any case, there is ample margin to take care of overload and a fairly definite general understanding as to what the rating signifies. * * *

With the steam turbine, on the other hand, there is no such general understanding and uniformity of practice, and it is not safe for the operating steam engineer to assume that a turbine with a given rating is capable of carrying the same overload that may be saddled upon the reciprocating engine of the same rating. * * *

* * * A uniform method of rating should be adopted just as quickly as possible and nobody is in a better position to determine what that method should be than The American Society of Mechanical Engineers. By adopting such rules concerning steam machinery as the Institute of Electrical Engineers has done for electrical machinery, it will perform a real service to the country. * * *

Carl Geo. de Laval. A new steam turbine has lately been designed on quite radical lines by Messrs. Ljungstrom, Stockholm, Sweden. It is a radial flow reaction turbine with steam entering between two disks passing from center to circumference between concentric blading rings. These two disks revolve with their shafts at the same rate of speed, but in opposite directions. At the end of each shaft an electric generator is attached. By doing this the relative speed of each blade becomes double the amount of the ordinary reaction turbine of the same number of revolu-

[1] October 14, 1911.

tion and diameter, with the consequent result that the blade rows become only one quarter as great. The general result is a turbine of small dimensions for the power and consequent smaller floor space even with the generators included than the present designs now on the market. Several turbines have been built and are being built.

The first turbine was 500 kw. at 3000 revolutions, which gave an efficiency value of 71.8 per cent and steam consumption 8.75 lb. per b.h.p. per hr., with steam of 175 lb. absolute and 250 deg. fahr. superheat and vacuum 28.5 in.

Sizes from 1000 to 7500 kw. with speed of 3000 revolutions have been designed on this double-rotating principle. Single rotating turbines of the same design have also been brought out, but with slightly less efficiency and increased steam consumption, the latter being about 10 per cent, comparing the recorded efficiency ratio given by the author. Erste Brünner M.F.G. on a Curtis-Parsons is the highest recorded at 71.8 per cent, steam 156.2 lb., 1500 revolutions, lb. of steam per kw-hr., 13.82. The Ljungstrom 1000-kw. steam turbine gave 74.7 per cent efficiency ratio, taking steam in front of the throttle valve and 76.9 per cent behind the throttle valve, steam 162 lb., speed 3000 revolutions, lb. of steam per kw-hr., 11.55 or 8 per b.h.p. It is stated that the critical speed of this turbine is 6300 revolutions, preventing the necessity as in many turbines of passing through the critical speed.

There are many novel details carried out in this turbine. The bottom is so shaped that it is placed directly on the condenser without the use of expansion joints. The two generators are supported on springs or slides on top of the extending ends of the condenser. The blades are of the Parsons type, milled from a solid bar of nickel steel, and having notched ends with blade roots turned over into the rings. In starting, the two sides of the turbine may not run at the same speed, but the exciting currents are strong enough so that at a certain speed the generators will synchronize automatically and run as one unit.

W. L. R. Emmet. In Table 2, which gives a comparison of various turbine performances, injustice is done to the Curtis type, and other incorrect impressions are given. This table is derived from various chance publications, but nothing is known concerning the authenticity of any of them. Such classification

by merit as Professor Christie has made therefore ought not to be published without qualification.

The first machine mentioned in this table is a 2000-kw. Curtis-Parsons operating at 1500 r.p.m. and is reported to give an efficiency of 71.8 per cent with a moderate degree of superheat. Everybody familiar with the design of either Curtis or Parsons machines knows perfectly well that no such result has ever been produced under such conditions.

The efficiencies assigned to the five General Electric Curtis

TABLE 3 TESTS OF REPRESENTATIVE CURTIS TURBINES

Customer	Date of Test	Load, Kw.	R.p.m.	Steam Pressure Absolute	Superheat, Deg. Fahr.	Vacuum	Water Rate, Lb. per Kw-Hr.	Efficiency, Per Cent
		7526	720	184.40	134	28.65	13.732	66.6
Boston Edison Co.	1907	7481	720	185.10	129	28.58	13.866	66.6
Commonwealth Edison Co., Chicago	1907	12000	750	200.00	125	28.00	14.220	66.2
Boston Edison Co.	1912	12460	720	207.50	191	27.52	13.630	68.2
Stock	1911	4000	1800	189.00	0	28.00	16.050	64.3
Louisville Ltg. Co.	1911	6500	1800	180.00	100	28.00	14.780	66.10
British Thomson - Houston, Rugby (maker)	1912	2000	3000	165.00	150	28.00	14.400	67.3

machines mentioned range from 63.6 per cent down to 61 per cent. The first, which operates at 3464 kw., is a representative result of the machine in question and is a good performance when the very high vacuum used is considered. Of the other four, only one of the tests is representative when the machine is correctly tested and in good condition, and in that case, also, the extremely high vacuum is the cause of a relatively low efficiency.

In discussing this table, the author states in Par. 70: "It is therefore apparent that the efficiency ratio alone will express in the best manner the degree to which the designer has approached ideal results in his turbine." To this statement exception should be taken for the reason that it is naturally possible to produce

higher efficiencies in turbine construction with moderate degrees of vacuum than it is with very high degrees of vacuum, the limitations to the effective use of high vacuum being many and their extent being affected by speed and capacity. Therefore the designer who produces a relatively low efficiency with a very high vacuum may have accomplished his purpose much more creditably than another who has produced a higher efficiency with a low vacuum.

None of the General Electric machines referred to in Professor Christie's tabulation, except the first mentioned, was designed later than 1904, while many of the other makes are of very recent production. Table 3 gives tests of a few representative Curtis machines. The first two tests apply to machines in Boston, very carefully and repeatedly tested when they were comparatively new. These machines are of the same date and type as most of those to which the figures in Professor Christie's tabulation purport to apply, and are representative of the original large 5-stage vertical units when correctly tested and in good condition. These tests were made with very perfect facilities by the Boston Edison Company's engineers.

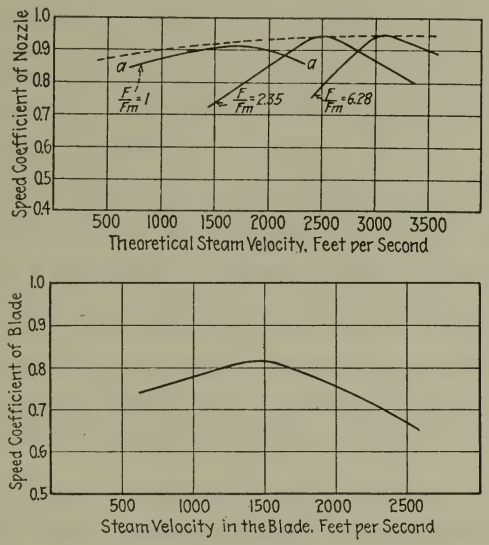
The third test applies to an 8000-kw. machine in Chicago of the same date and type. One of the tests in Professor Christie's paper also purports to refer to this machine under different conditions.

The next three items refer to recent six-stage Curtis machines, and the last refers to a three-stage Curtis machine recently produced by the British Thomson-Houston Company. This latter machine produces a very remarkable result when its small capacity and extreme simplicity are taken into consideration.

The accurate testing of steam units is a matter which requires great care and is susceptible of many kinds of error. Many figures concerning such tests which are entirely incorrect are being constantly circulated. It is only by comparison of many tests and careful observance of consistency as to results and characteristics that one can ever be sure as to turbine economies. The figures given in Table 3 have been analyzed and compared and are unquestionably correct. It will be observed that they give an impression concerning the value of Curtis turbines which is very different from that produced by Professor Christie's tabulation.

E. Heinrich[1] called attention to the importance of international coöperation in the development of the steam turbine. The Parsons turbine, originally emanating from England, has been developed and improved in America, while the Curtis principle, of American design, has been taken up in Europe by the Allgemeine Elektricitäts Gesellschaft, and developed so as to meet the requirements of the continental market.

Regarding the economic influence of steam velocities, it has for a long time been known that the specific losses in nozzles and



Figs. 14 and 15 Relation Between Theoretical Steam Velocity and Speed Coefficient

blades are essentially a function of these velocities, in such a manner that as the steam velocities increase, the specific losses decrease and the speed coefficient, i.e. the ratio of the actual to the theoretical velocities, increases. In practice advantage of this fact was taken by admitting higher steam velocities and thus reducing the number of stages, as is done, in the writer's opinion, in the Allis-Chalmers and Westinghouse designs in America and the new Zoelly design of 1909 in Europe.

The investigations of Professor Josse of Charlottensburg and others have revealed a relation between theoretical steam velocity

[1] Engrg. Laboratories, Royal Technical High School, Stuttgart, Germany.

and the speed coefficient shown in Figs. 14 and 15, Fig. 14 referring to nozzles of different divergencies, and Fig. 15 to running blades. The curves show that for every form of nozzle there is a most economical steam velocity which increases (as does the speed coefficient) with the increase of the ratio of outlet area to throat area of nozzle. This proves that a divergent nozzle, if properly designed, may have a better efficiency than a nozzle with parallel walls. Further, the curve *aa* relating to a nozzle of the latter type, shows that, contrary to former views, the economical steam velocity in nozzles of this type lies beyond the velocity of sound. These tests have also shown (Fig. 15) that the almost economical velocity in running blades nearly equals the velocity of sound. From this it would appear that the most advantageous number of stages in impulse turbines of the Rateau type is six, but in actual practice this number varies from 8 to 20.

One of the most important conclusions drawn from these tests is that referring to the efficiency of marine turbines. Being directly connected to the propeller shaft, their blade speed and consequently steam velocity are limited to a rather low range, with the result that they will always be inferior in economy to high-speed stationary turbines. Hence the increasing importance of reduction gears for large power transmissions, such as the Westinghouse reduction gear and the Foettinger Hydraulic Transformer.

As regards the turbine shaft, Professor Christie says that it is generally designed so that the normal speed should be considerably below the critical speed. This holds true for the drums of reaction turbines which are rigid on account of their construction, but in impulse turbines some European builders have adopted the use of the flexible shaft for the high-speed (3000 r.p.m.) types up to and over 3000 kw. From the commercial standpoint the application of flexible shafts has the following advantages: (*a*) the shaft and all parts depending on it are considerably smaller in diameter than when a rigid shaft is used, and therefore the first cost is less; for instance, in the case of a Rateau type 2000-kw. turbine, with 3000 r.p.m. and 9 stages, a flexible shaft of about 6 in. medium diameter is required, the critical speed being about 2000 r.p.m. If a rigid shaft is used, its diameter has to be about 10 in., and the critical speed up to 4000 r.p.m. (the critical and normal speed must differ widely in

order to prevent their coincidence); (*b*) with respect to the steam economy, the flexible shaft is of advantage because the diameters of the stuffing boxes at the guide disk hubs are smaller than when a rigid shaft is used. Hence the leakage losses between stationary and rotary parts is smaller with flexible shafts than with rigid shafts.

THE AUTHOR. In the discussion it has been held that as high an efficiency ratio cannot be obtained with high vacuum and low superheat as with moderate vacuum and high superheat. This is probably true of the Curtis turbine, since apparently it does not utilize high vacuum as economically as other types, which was pointed out in the paper itself. This is therefore a characteristic of only this type of turbine.

If a turbine is designed for a large amount of available energy there is no reason why fully as high an efficiency ratio should not be obtained as with a small amount of available energy. Sufficient data are not at hand to warrant a general statement for all types that such conditions as above referred to cannot be fulfilled.

In regard to the increasing use of the efficiency ratio as a measure of steam turbine economy, it is interesting to note that a leading European firm includes a statement in their steam turbine specifications to the effect that should the steam or vacuum conditions at the time of the tests vary from those specified, the efficiency ratio of the unit shall not be less than that represented by the guarantee.

The author recognizes that the efficiency ratio alone does not provide all the information desirable for a given turbine, but when used in conjunction with specified steam and vacuum conditions, it provides the most reliable index to the actual performance of the steam turbine.

It is very gratifying to note the attention given to the present uncertain state of steam turbine ratings. It is to be hoped that the Committee on Power Tests of the Society will bring in an early report on this subject and fix certain standards of rating.

Objections have been raised to the Erste Brünner tests. Before including these in Table 2, inquiry was made as to their reliability. These tests were conducted by Prof. K. Koerner, one of the most prominent of Austrian engineers, who substantiated the figures quoted.

Mr. H. Zoelly presented a paper[1] at the 1911 meeting in Zurich, Switzerland, of the Institution of Mechanical Engineers in which he made the statement quoted in Par. 51. He gave his reasons for such an opinion and also presented results to verify his statements. An abstract of this paper was printed in Engineering, April 29, 1911, and can be referred to by those interested.

Mr. Crissey presents a very excellent classification of steam turbines. However, for the purpose of this paper, the author preferred to classify turbines not so much as pure theoretical types, but under a classification of names and types with which engineers in general are familiar, hoping thereby to avoid confusion.

There seems to be a diversity of opinion regarding blade erosion in Curtis turbines. Some users have reported serious erosion in much less time than Mr. Moyer quotes. These difficulties are being steadily overcome by improvements in design.

Regarding Mr. Hodgkinson's statement concerning blade troubles, the author fails to recall in his own experience, under ordinary conditions of operation, a case of blade failure due to centrifugal force alone.

The value of test results supplied by manufacturers has been criticised. There have been cases in the past where such results have been questionable, but in general manufacturers now find it advisable to publish only such tests as represent the actual and usual performance of their machines. It is frequently the case that steam turbines sold subsequent to the publication of such tests must operate with the same degree of economy as represented by the test results.

The corrections for Table 1 given by Mr. Hodgkinson are very welcome, bringing the data presented strictly up to date. The description of the new type of Westinghouse blading is also of considerable interest to engineers.

In connection with Dr. Stodola's tests of a Sulzer turbine quoted by Mr. Moyer, it was the author's impression when visiting Sulzer's works that while their Diesel engines probably excelled all other makes, their steam turbines were not representative of the best advanced practice in Curtis-Parsons construction, owing to characteristics of general design and particularly to their Parsons blading. This blading was much heavier in section than the usual form, and was not shrouded or otherwise re-

[1] Steam-Turbines, Proc. I. Mech. Engrs., 1911, No. 3, p. 681.

TABLE 4 ECONOMY TESTS OF HIGH PRESSURE STEAM TURBINES

Efficiency Ratios based on E. H. P.

Maker of Turbine	Type	Date of Test	Load, Kw.	R.P.M.	Steam Pressure, Lb. Absolute	Temperature at Throttle, Deg. Fahr.	Vacuum Referred to 29.92 In. Barometer	Condenser Pressure, Lb. Absolute	Lb. of Steam per Kw-Hr.	B.t.u. per Kw-Hr.	Heat Utilized per Lb. of Steam	Heat Available per Lb. of Steam	Efficiency Ratio	Reference
Escher Wyss & Co.	Zoelly	1912	1275	3000	196.3	588	28.99	0.455	12.59	15940	268.6	418.1	64.3	Zeit. d. v. d. Ing., 4 13 '12
Escher Wyss & Co.	Zoelly	1912	5418	1500	180.7	588	28.74	0.580	11.95	15060	285.5	402.1	71.0	
Elsassische Maschinenbau Gesellschaft	Zoelly	1912	10006	1250	167.8	565	28.65	0.625	12.01	14975	284.2	389.1	73.0	
Belliss & Morcom	Drum Impulse	1911	3000	1500	192.5	558	27.92	0.981	15.0	18420	227.3	372.7	61.0	Manufacturers' Data
Belliss & Morcom	Drum Impulse	1911	2500	1500	164.7	466	26.92	1.471	16.95	19815	201.2	323.6	62.2	
Aktiebolaget Ljungström Angturbin...	Radial Reaction (Ljungström)	1911	1000	3000	181.7	669	28.55	0.671	11.68	15130	292.0	414.9	70.4	Engineering, 4 19 '12
		1911	972	3000	176.7	667	28.58	0.657	11.55	14960	295.3	414.2	71.3	

Marks and Davis tables used.

inforced. The tests quoted amply bear out this impression, and the author believes that such results should not be considered as typically representative of the whole class of Curtis-Parsons turbines.

The test results presented by Mr. Emmet for Curtis turbines show decided improvements in design over the earlier Curtis turbines in Table 2, but still do not exceed the Curtis-Parsons or Curtis-Rateau results. Table 4 presents some results of tests which were available subsequent to the preparation of Table 2, and which show some remarkable performances of Zoelly turbines. The Bellis and Morcom tests were on some of their earlier machines, and it is reasonable to expect considerable improvement in units now under construction. The results obtained by the Ljungström turbines are remarkable for high efficiency. The design of the details of this turbine shows a high degree of originality on the part of the inventor. The turbine is said to be very cheap to build, but before putting it on the American market many parts would have to be simplified and improved, particularly the bearings, steam packing and blade holding rings.

No. 1356

A DISCUSSION OF CERTAIN THERMAL PROPERTIES OF STEAM

By G. A. Goodenough,[1] Urbana, Ill.

Non-Member

The recent paper [2] by Dr. H. N. Davis on the thermal properties of steam covers the ground so completely that any further contribution may seem superfluous. In one phase of the investigation, however, he acknowledges some lack of success and leaves unsolved a single problem. He says on page 303, "The reconciliation through Clausius' thermodynamic relation of the accepted volume and specific heat measurements in the superheated region is impossible. It is probably the most important of the outstanding problems in the field." It is the purpose of this paper to present an attack on the problem thus suggested.

2 The Clausius relation referred to is expressed by the equation

$$\left[\frac{\partial c_p}{\partial p}\right]_T = -AT\left[\frac{\partial^2 v}{\partial T^2}\right]_v$$

The relation is readily derived from the general equations of thermodynamics, and is applicable to all substances. The derivative in the second member is obtained from the characteristic equation $\phi\ (p, v, t) = 0$ of the substance under consideration. If an expression can be found for this derivative from a sufficiently accurate characteristic equation, the rate of variation of specific heat c_p with the pressure is determined; and if sufficient experimental data are available, it should be possible to derive an explicit expression for c_p in terms of the variables p and T. The method has inherent difficulties, as

[1] Professor of Thermodynamics, Univ. of Ill.

[2] Notes on Certain Thermal Properties of Steam, H. N. Davis. Proc. Am. Acad. of Arts and Sciences, vol. 45, pp. 268–311.

Presented at the Spring Meeting, Cleveland, 1912, of The American Society of Mechanical Engineers, 29 West 39th Street, New York.

pointed out by Dr. Davis, and so far it has been used with little success.

3 In the following sections an attempt is made to apply the Clausius relation in the case of superheated steam and to reconcile the volume measurements of Knoblauch, Linde, and Klebe with the specific heat measurements of Knoblauch and Jakob, and of Knoblauch and Mollier. The discussion is divided as follows:

a The characteristic equation for superheated steam proposed by Linde is modified without any material sacrifice in accuracy.

b From the modified equation an expression for the specific heat c_p is derived, and the constants are so chosen that the specific heat curves represent quite accurately the results obtained by Knoblauch and Jakob, and Knoblauch and Mollier.

c From the equation for c_p thus developed, explicit equations for the heat content, entropy and energy of superheated steam are derived.

d The values of volume, heat content and entropy deduced from these equations are compared with the values given in the Marks and Davis tables, and substantial agreement is shown.

THE CHARACTERISTIC EQUATION

4 The experiments of Knoblauch, Linde and Klebe furnish reliable data for a characteristic equation connecting the pressure, temperature and volume of superheated steam. To represent these experiments, Linde [1] proposed in the first instance the equation

$$pv = BT - \left[C\left(\frac{373}{T}\right)^2 - D\right]\frac{1}{v} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

but for the sake of convenience in calculation adopted finally the equation

$$pv = BT - p(1+ap)\left[C\left(\frac{373}{T}\right)^3 - D\right] \quad \ldots\ldots\ldots\ldots\ldots [2]$$

Linde gives for the constants of [2] the following values (metric units):

$$B = 47.10;\ a = 0.000002;\ C = 0.031;\ D = 0.0052$$

[1] Mitteilungen über Forschungsarbeiten, vol. 21, 1905.

It appears, however, that these constants can be changed somewhat without affecting materially the agreement of the calculated and experimental results. Linde himself calls attention to an objection to the assumed form of the equation which lies in the fact that at a temperature of about 402 deg. cent., the "correction term" of the second member changes sign, and for higher temperatures the fluid becomes a "more than perfect" (übervollkommenes) gas. He suggests in this connection that the objection may be obviated

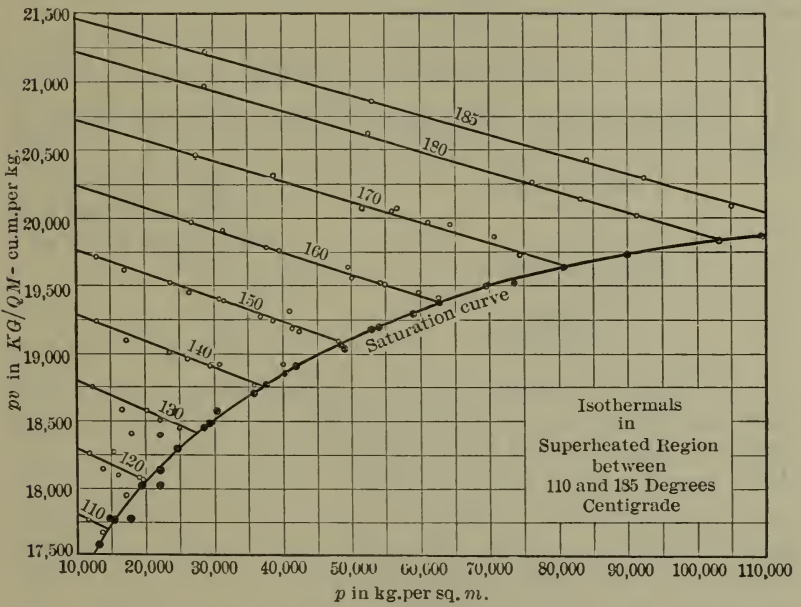


FIG. 1 COMPARISON OF ISOTHERMAL CURVES DEDUCED FROM [3] WITH THE POINTS REPRESENTING THE EXPERIMENTS OF KNOBLAUCH, LINDE AND KLEBE

by increasing the exponent of T from 3 to 3.5, the value assumed by Callendar. The integral value 3 was retained, however, for the sake of convenience in computation.

5 Close inspection of Linde's equation shows that its form is such as to render it somewhat inconvenient for purposes of computation. The essential feature of the formula is the factor $(1+ap)$ which gives the parabolic form to the isothermals when drawn on the pv-p plane. If this be retained, considerable liberty may be taken with the remaining constants without materially affecting the integrity of the equation. In the first place, if the exponent n

be raised from 3 to 3.5, the constant D may be dropped entirely. The equation is then free from Linde's objection that the correction term changes sign at high temperatures, and with constants properly chosen gives results that represent satisfactorily the Munich experiments.

6 In the investigation of other properties of superheated steam, heat content, entropy, etc., considerations appeared that rendered desirable further modification of Linde's equation. The exponent n was raised to 5 and a constant was added to the volume v. The equation therefore took the form:

$$p(v+c)=BT-p(1+ap)\frac{m}{T^n} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

7 The following are the constants finally chosen for this equation:

Metric Units	English Units
$B=47.113$	$B=85.871$, p in lb. per sq. ft. $=0.5963$, p in lb. per sq. in.
$\log m=11.19839$	$\log m=13.67938$
$n=5$	$n=5$
$c=0.0055$	$c=0.088$
$a=0.00000085$	$a=0.0006$, p in lb. per sq. in.

8 With these constants equation [3] represents the results of the Munich experiments with substantially the same accuracy as Linde's equation. In fact, when the isothermal curves are drawn on a pv-p chart, as in Fig. 13 of Linde's discussion, the agreement between the curves and experimental points is in some respects better than that given by Linde's equation. The isothermals thus drawn are shown in Fig. 1, which may be compared with Linde's Fig. 13. For a further test of equations [2] and [3], showing that the latter equation may be accepted with confidence at the saturation limit, see Appendix No. 1.

9 As the experiments of Knoblauch, Linde and Klebe extended to only 185 deg. cent., the validity of [3] for higher temperatures is uncertain. It is a safe assertion, however, that [3] is more suitable for extrapolation to high temperatures than is the original Linde equation, but it is possible that equation [3] does not hold at very low temperatures (0 deg. cent.-40 deg. cent.) in the vicinity of the saturation curve.

FORMULA FOR SPECIFIC HEAT

10 The application of the Clausius relation to equation [3] readily gives the following equation for the specific heat c_p:

$$c_p = \phi(T) + \frac{Amn(n+1)}{T^{n+1}} p\left(1+\frac{ap}{2}\right) \quad \ldots\ldots\ldots\ldots [4]$$

or

$$c_p = \phi(T) + f(p, T) \quad \ldots\ldots\ldots\ldots\ldots\ldots [4a]$$

if the term $\frac{Amn(n+1)}{T^{n+1}} p\left(1+\frac{ap}{2}\right)$ is replaced for convenience by $f(p, T)$. For the complete derivation of this equation see Appendix No. 2.

11 If the characteristic equation [3] is correct in form, the expression for the specific heat c_p must have the form given by [4]; and if the arbitrary function ϕ (T) can be determined, we shall have an explicit expression for c_p in terms of the variables p and T. Callendar and Linde have both used this analysis, starting from their respective characteristic equations, and have thus obtained equations similar to [4].

12 The determination of the proper form of the function ϕ (T) is the critical point of the whole investigation. If in [4], $p=0$, the equation reduces to $c_p = \phi$ (T). Callendar reasons that at zero pressure the specific heat c_p should have an invariable value $(c_p)_o$ independent of the temperature; hence $\phi(T)$ is assumed to be constant. Linde follows Callendar in this assumption. Knoblauch and Jakob, however, on the strength of their direct experiments on the values of c_p, show that $(c_p)_o$ cannot be a constant. It seems possible to establish the form of the function by quite another line of reasoning. Inspection of [4] shows that the correction term f (p, T), having the factor T^{n+1} in the denominator, grows rapidly smaller as T increases. At moderately low temperatures ($T=100$ to 600 deg. cent.) this term has an appreciable value, but at temperatures above say 1000 deg. cent., the exponent $n+1$ being 6, the term practically disappears, and the specific heat is given by the relation

$$c_p = \phi\ (T) \quad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

for all pressures. The determination of the function $\phi(T)$ is reduced therefore to the discovery of the relation between specific heat and temperature at high temperatures. The experiments of Mallard and Le Chatelier at temperatures above 3000 deg. cent. and the experiments of Langen at about 1700 deg. cent. agree in making the specific heat a linear function of the temperature, for the temperature range within which the experiments were made. While this linear relation may not, and probably does not, hold for the entire region of superheat, it may be assumed as a close approxi-

mation for a moderate range of temperature. The function $\phi(T)$ therefore takes the simple form

$$\phi(T) = \alpha + \beta T \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [6]$$

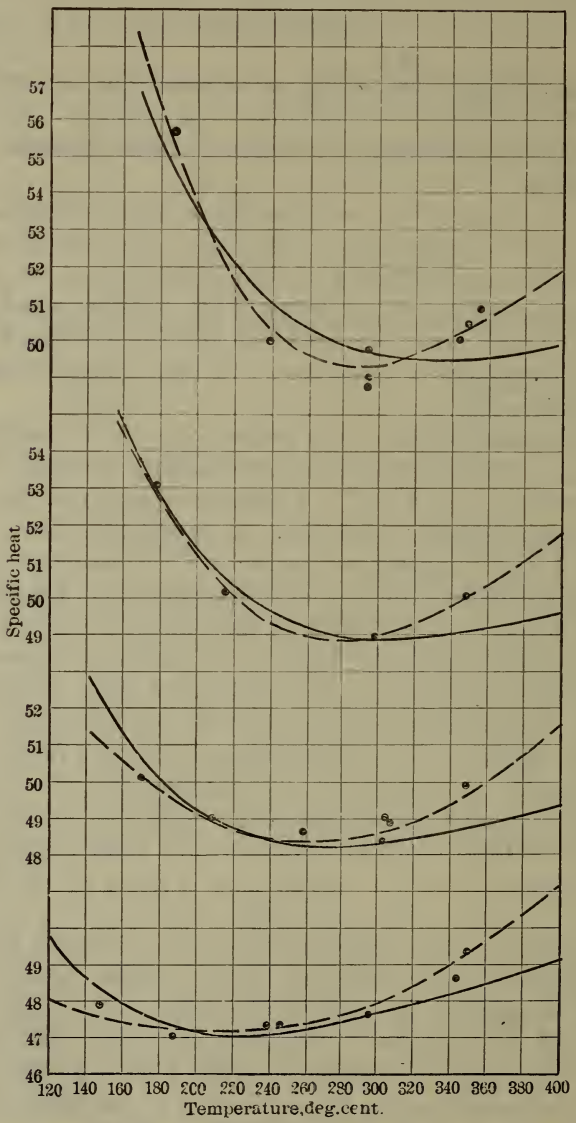


FIG. 2 COMPARISON OF SPECIFIC HEAT CURVES WITH KNOBLAUCH AND JAKOB POINTS. PRESSURES, 2, 4, 6, AND 8 KG. PER SQ. CM.

and equation [4] becomes

$$c_p = \alpha + \beta T + \frac{Amn(n+1)}{T^{n+1}} p\left(1 + \frac{ap}{2}\right) \quad \ldots\ldots\ldots\ldots [7]$$

13 The constants a, m, and n are those heretofore given in connection with the characteristic equation [3], while in the determination of the constants α and β we may utilize the results of the various experiments of Knoblauch and Jakob, and the more recent experiments of Knoblauch and Mollier. For a fuller discussion of the values of these constants see Appendix No. 3.

14 The values finally chosen were:

$$\alpha = 0.367 \qquad \beta = 0.00018$$

Hence

$$\phi\,(T) = 0.367 + 0.00018T \ldots\ldots\ldots\ldots\ldots\ldots [8]$$

or for the fahrenheit scale

$$\left.\begin{aligned} \phi\,(T) &= 0.367 + 0.0001T \\ \phi\,(T) &= 0.413 + 0.0001t \end{aligned}\right\} \ldots\ldots\ldots\ldots\ldots\ldots [9]$$

The agreement between the curves that represent equation [7], using the expression for $\phi\,(T)$ given by [8], and the Knoblauch and Jakob points is shown in Fig. 2. The agreement is quite satisfactory except for points in the region of 350 deg. The empirical curves adopted by Knoblauch and Jakob (shown in dash lines) were made to pass through these points and were therefore given a sharp slope above 350 deg. Furthermore, at the higher pressures the empirical curves were given much greater slopes at the intersections with the saturation curve than were shown by the curves calculated from [7]. As a result the extrapolated values of c_p at the saturation limit are suspiciously high. Marks and Davis, on the strength of results of Holborn and Henning, lowered the Knoblauch and Jakob curves in the region of high superheat, but accepted their values near saturation.

15 The experiments of Knoblauch and Mollier extend the temperature range to 550 deg. cent. The points representing these experiments are shown in Fig. 3, and the curves exhibited in the same figure represent equation [7] with the values of α and β originally chosen. For the lower pressures the agreement between points and curves is satisfactory. At the higher pressures the agreement could be slightly improved by raising the value of α from 0.367 to 0.369 or 0.370. For further discussion of the validity of equation [7] and values chosen for the constants α and β see Appendix No. 6.

HEAT CONTENT, ENERGY AND ENTROPY

16 With the aid of an explicit formula for the specific heat, the following expressions for the heat content, energy and entropy of superheated steam are readily derived:

For heat content:

$$i = aT + \tfrac{1}{2}\beta T^2 - \frac{Am(n+1)}{T^n} p\left(1+\frac{ap}{2}\right) - Acp + i_o \ldots\ldots\ldots [10]$$

For intrinsic energy (in B.t.u.):

$$u = T(a + \tfrac{1}{2}\beta T - AB) - \frac{Amp}{T^n}\left[n + (n-1)\frac{ap}{2}\right] + i_o \ldots\ldots\ldots [11]$$

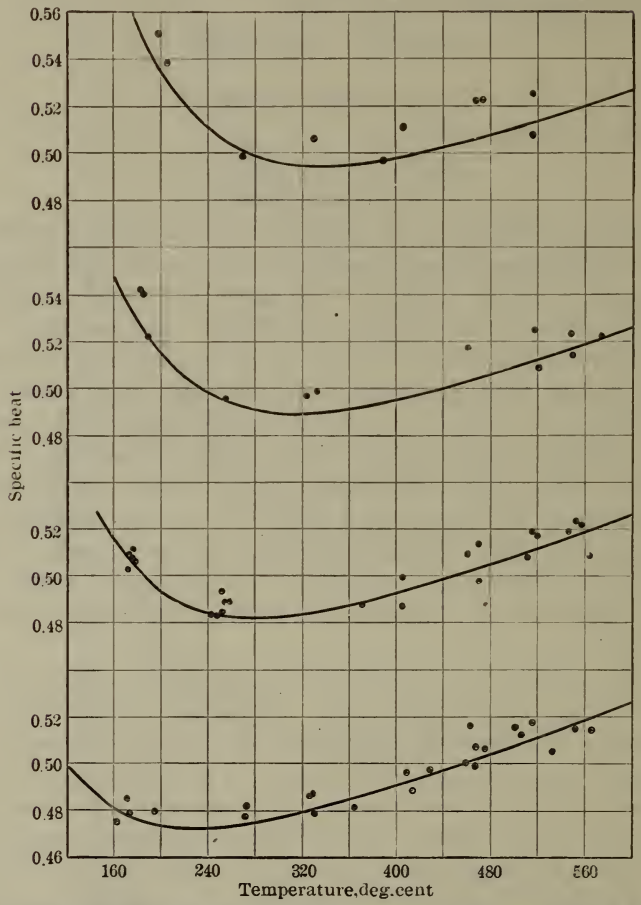


FIG. 3 COMPARISON OF SPECIFIC HEAT CURVES WITH KNOBLAUCH AND MOLLIER POINTS. PRESSURES, 2, 4, 6, AND 8 KG. PER SQ. CM.

For entropy:

$$s = a \log_e T + \beta T - AB \log_e p - Anp\left(1+\frac{ap}{2}\right)\frac{m}{T^{n+1}} + s_o \ldots\ldots [12]$$

For the derivation of these equations see Appendix No. 4.

17 Formula [10] gives the heat content i of superheated steam at any assumed pressure p and temperature T. Since the formula should hold good at the saturation limit, it also gives the value of i for saturated steam when corresponding saturation values of p and T are used. A comparison of the saturation values of i thus obtained with the values given by Marks and Davis will furnish a test of the accuracy of equation [10], and incidentally a test of the entire theory built up from equations [3] and [7]. Taking 1150.4 B.t.u. as the value of i for saturated steam at 212 deg. (the value given by Marks and Davis), the constant i_o is found to be 887.3; and using this value of i_o, the values of i shown in Table 1 are obtained. For a further discussion of the agreement between the two sets of values, and in particular for a proposed modification of Dr. Davis' formula for the heat content see Appendix No. 5.

TABLE 1 VALUES OF i AT SATURATION

Temperature, Deg. Fahr.	Calculated from [10], $i_o = 887.3$	Marks and Davis	Difference, Per Cent
32	1079.6	1073.4	0.58
50	1087.0	1081.4	0.52
75	1097.3	1092.5	0.44
100	1107.4	1103.6	0.34
125	1117.4	1114.5	0.26
150	1127.2	1125.3	0.17
200	1146.0	1145.8	0.02
212	1150.4	1150.4	0.00
250	1163.5	1163.8	−0.03
300	1179.5	1179.1	+0.04
350	1193.6	1191.5	0.18
400	1205.7	1201.3	0.35
450	1215.3	1208.0	0.61

18 From the comparison of the two sets of values in Table 1, and from the considerations set forth in Appendix No. 5, the following conclusions may be drawn:

(*a*) Equation [10] gives with fair accuracy the saturation values of the heat content from 120 deg. to 450 deg. fahr., that is, from a pressure of 1.7 lb. per sq. in. to 420 lb. per sq. in.

(*b*) Equation [10] therefore probably gives quite accurately the heat content of superheated steam for this pressure range and for the degree of superheat usually employed in technical applications.

(*c*) The validity of equation [10] carries with it the validity of the analysis, and justifies the choice of constants in equations [3] and [7].

(*d*) For extremely low pressures the exact values of the various properties of steam are still in doubt, and further investigation is required.

RÉSUMÉ OF FORMULAE

19 The formulae for the important properties of superheated steam, with the numerical values of the constants inserted, are here collected for convenience of reference. In all cases the constants are so chosen that pressures are to be taken in pounds per square inch:

(*a*) For the specific volume:

$$v = 0.5963\frac{T}{p} - (1+0.0006p)\frac{C}{T^5} - 0.088$$

$$\log C = 13.67938$$

(*b*) For the specific heat:

$$c_p = 0.367 + 0.0001T + p(1+0.0003p)\frac{C_2}{T^6}$$

$$\log C_2 = 14.42408$$

(*c*) For the heat content:

$$i = T(0.367+0.00005T) - p(1+0.0003p)\frac{C_3}{T^5} - 0.0163p + 887.3$$

$$\log C_3 = 13.72511$$

(*d*) For the intrinsic energy in B.t.u.:

$$u = T(0.2566+0.00005T) - p(1+0.00024p)\frac{C_4}{T^5} + 887.3$$

$$\log C_4 = 13.64593$$

(*e*) For the entropy:

$$s = 0.8451 \log T + 0.0001T - 0.2542 \log p - p(1+0.0003p)\frac{C_5}{T^6} - 0.3960$$

$$\log C_5 = 13.64593$$

COMPARISON OF RESULTS

20 From the formulae in Par. 19 the properties of superheated steam may be calculated for assumed values of p and T. It may be of interest to compare the values thus obtained with the values given in the Marks and Davis table. Such a comparison is shown in Table 2. Various pressures from 2 to 300 lb. per sq. in. are assumed and the superheat ranges from 0 deg. to 400 deg. fahr.

TABLE 2 COMPARISON OF THE CALCULATED VALUES OF THE PROPERTIES OF SUPERHEATED STEAM WITH THE VALUES GIVEN IN THE MARKS AND DAVIS TABLE

	Pressure, Lb. per Sq. In.														Superheat, Deg.
	2		15		50		100		150		200		300		
	Cal.	(M and D)	Cal.	(M and D)	Cal.	(M and D)	Cal.	(M and D)	Cal.	(M and D)	Cal.	(M and D)	Cal.	(M and D)	
Volume, v			26.32												
	174.0	173.5	26.32	26.27	8.52	8.51	4.44	4.43	3.02	3.01	2.30	2.29	1.55	1.55	0
	189.1	189.2	28.41	28.40	9.18	9.19	4.79	4.79	3.26	3.27	2.48	2.49	1.67	1.69	50
	204.2	204.2	30.47	30.46	9.83	9.84	5.12	5.14	3.49	3.51	2.65	2.68	1.79	1.83	100
	234.2	234.2	34.53	34.53	11.07	11.11	5.75	5.80	3.92	3.97	2.98	3.04	2.01	2.09	200
	294.0	293.9	42.55	42.59	13.51	13.58	6.98	7.07	4.74	4.83	3.60	3.71	2.43	2.55	400
Heat Content, i															
	1117.8	1115.0	1150.7	1150.7	1173.6	1173.6	1187.6	1186.3	1195.9	1193.4	1201.6	1198.1	1209.3	1204.1	0
	1139.8	1137.8	1174.3	1174.2	1199.2	1198.8	1214.8	1213.8	1224.2	1223.0	1231.0	1229.8	1240.2	1240.3	50
	1161.7	1160.7	1197.3	1197.6	1223.9	1223.4	1240.8	1239.7	1251.2	1249.6	1258.7	1257.1	1269.2	1268.2	100
	1206.1	1206.4	1243.8	1244.4	1272.3	1271.8	1291.0	1289.4	1302.7	1300.0	1317.3	1307.7	1323.7	1319.3	200
	1297.5	1298.6	1337.7	1338.3	1368.7	1367.4	1389.6	1385.9	1403.0	1397.4	1412.9	1405.9	1427.6	1418.6	400
Entropy, s															
	1.9231	1.9180	1.7549	1.7549	1.6578	1.6581	1.6028	1.6020	1.5710	1.5692	1.5490	1.5456	1.5175	1.5129	0
	1.9587	1.9559	1.7887	1.7886	1.6911	1.6909	1.6363	1.6358	1.6049	1.6043	1.5828	1.5823	1.5518	1.5530	50
	1.9921	1.9904	1.8199	1.8199	1.7214	1.7211	1.6665	1.6658	1.6350	1.6343	1.6130	1.6120	1.5822	1.5824	100
	2.0526	2.0529	1.8761	1.8768	1.7759	1.7755	1.7201	1.7188	1.6884	1.6862	1.6662	1.6632	1.6354	1.6323	200
	2.1561	2.1586	1.9730	1.9733	1.8688	1.8671	1.8111	1.8079	1.7781	1.7735	1.7555	1.7493	1.7238	1.7168	400

21 It will be seen that there is substantial agreement between the values for the heat content i and also between the values for the entropy s. The maximum difference between the values of i is slightly over 0.6 of one per cent and that between the values of s is about 0.4 of one per cent. The discrepancy between the values of v is somewhat larger at the higher superheats. From Linde's own statement of the defect in his formula when applied at high temperatures, it appears probable that the values obtained from formula [3] are more accurate than those calculated from Linde's formula. It is remarkable that the values of i and s obtained from quite different specific heat curves should agree so well. And it is gratifying that the accuracy of the Marks and Davis tables is confirmed by this close agreement.

22 Acknowledgment is due Mr. A. L. Schaller and Mr. John A. Dent for assistance in the calculations and for valuable suggestions.

APPENDIX No. 1

VALIDITY OF EQUATIONS [2] AND [3] AT THE SATURATION LIMIT

$$pv = BT - p(1+ap)\left[C\left(\frac{373}{T}\right)^3 - D\right]$$

$$p(v+c) = BT - p(1+ap)\frac{m}{T^n}$$

23 Equations [2] and [3] may also be tested by calculating from them values of v at the saturation limit and comparing these with values determined experimentally or with values calculated from the properties of saturated steam by means of the well known Clapeyron-Clausius formula. Table 3 gives a series of values thus determined.

TABLE 3 SPECIFIC VOLUME v AT SATURATION

Temperature, Deg. Cent.	Experiments of K., L. and K.	Calculated from [2]	Calculated from [3]	Calculated by Henning
100	1.674	1.674	1.673	1.673
120	0.8922	0.8915	0.8915	0.8912
140	0.5091	0.5083	0.5084	0.5078
160	0.3073	0.3070	0.3071	0.3071
180	0.1943	0.1943	0.1945	0.1947

24 The values given in the last column are of special significance. They are those calculated by Henning, using as a basis his recently determined values of the latent heat and Holborn and Henning's accurate values of $\frac{dp}{dt}$. That these values agree so closely with the values determined by the direct experiments of Knoblauch, Linde and Klebe is convincing evidence of the accuracy attained in both sets of experiments. It will be observed that the values of v_{sat} given by [3] agree substantially with both sets of values; hence it appears that equation [3] may be accepted with confidence at the saturation limit.

APPENDIX No. 2

DERIVATION OF EQUATION [4]

$$c_p = \phi(T) + \frac{Amn(n+1)}{T^{n+1}} p\left(1+\frac{ap}{2}\right)$$

25 Writing the characteristic equation [3] in the form

$$v+c = \frac{BT}{p} - (1+ap)\frac{m}{T^n}$$

the following derivatives are readily obtained:

$$\frac{\partial v}{\partial T} = \frac{B}{p} + \frac{mn}{T^{n+1}}(1+ap) \quad \ldots\ldots [13]$$

$$\frac{\partial^2 v}{\partial T^2} = -\frac{mn(n+1)}{T^{n+2}}(1+ap) \quad \ldots\ldots [14]$$

Substituting the second derivative in the Clausius equation

$$\left[\frac{\partial c_p}{\partial p}\right]_T = -AT\frac{\partial^2 v}{\partial T^2}$$

the result is the relation

$$\left[\frac{\partial c_p}{\partial p}\right]_T = \frac{Amn(n+1)}{T^{n+1}}(1+ap) \quad \ldots\ldots [15]$$

Taking T as a constant and integrating [15] with p as the independent variable, the result is

$$c_p = \frac{Amn(n+1)}{T^{n+1}}\ p(1+\frac{ap}{2}) + \text{constant of integration}$$

The constant of integration may be a function of T, since T was held constant during the integration; hence

$$c_p = \phi(T) + \frac{Amn(n+1)}{T^{n+1}} p(1+\frac{ap}{2}) \quad \ldots\ldots [16]$$

APPENDIX No. 3

CONSTANTS α AND β IN EQUATIONS [6] AND [7]

$$\phi(T) = \alpha + \beta T$$

$$c_p = \alpha + \beta T + \frac{Amn(n+1)}{T^{n+1}} p\left(1 + \frac{ap}{2}\right)$$

26 The limits of the value of β are furnished by the linear equations deduced from the explosion experiments and from the experiments of Holborn and Henning. Thus the results of the experiments of Mallard and Le Chatelier at a temperature above 3000 deg. are represented by the equation

$$c_p = 0.432 + 0.000318T$$

Langen's experiments at about 1700 deg. cent. are represented by the equation

$$c_p = 0.438 + 0.000239T$$

and the experiment of Holborn and Henning for temperature below 800 deg. cent. by the equation

$$c_p = 0.446 + 0.0000956T$$

There is reason to believe that the values given by Holborn and Henning are too low. It appears that the coefficient of T is not constant but increases with the temperature; and the value chosen for β will be influenced by the temperature range over which the formula for c_p is assumed to extend. Since in technical applications the temperatures encountered are relatively low, β should probably be smaller than Langen's value 0.000239, and bearing in mind the probable error in the results of Holborn and Henning, β should be considerably higher than the value 0.0000956.

27 In the original determination of the constants α and β, the results of the experiments of Knoblauch and Jakob were used. The later experiments of Knoblauch and Mollier had not been published. The "correction term" $f(p, T)$ of equation [7] was calculated for assumed temperatures and for the pressures 2, 4, 6 and 8 kg. per sq. cm. employed by Knoblauch and Jakob. Then by trial, values of α and β were so chosen as to make the specific heat curves

$$c_p = \alpha + \beta T + f(p, T)$$

fit as closely as possible the experimental points.

APPENDIX No. 4

DERIVATION OF EQUATIONS [10], [11] AND [12]

$$i = aT + \tfrac{1}{2}\beta T^2 - \frac{Am(n+1)}{T^n} p\left(1 + \frac{ap}{2}\right) - Acp + i_o$$

$$u = T(a + \tfrac{1}{2}\beta T - AB) - \frac{Amp}{T^n}\left[n + (n-1)\frac{ap}{2}\right] + i_o$$

$$s = a \log_e T + \beta T - AB \log_e p - Anp\left(1 + \frac{ap}{2}\right)\frac{m}{T^{n+1}} + s_o$$

28 With the aid of an explicit formula for the specific heat, expressions for the heat content, energy, and entropy of superheated steam are as easily derived. For this purpose the general equation

$$dq = c_p dT - AT\left(\frac{\partial v}{\partial T}\right)_p dp \qquad [17]$$

is most convenient. Since the heat content i is defined by the relation

$$i = A(u + pv)$$

we have

$$di = c_p dT - A\left(T\frac{\partial v}{\partial T} - v\right)dp \qquad [18]$$

From the characteristic equation [3] the derivative $\frac{\partial v}{\partial T}$ is found to be

$$\frac{\partial v}{\partial T} = \frac{B}{p} + \frac{mn}{T^{n+1}}(1 + ap)$$

Substituting this and also the expression for c_p in [18], the result is

$$di = (a + \beta T)dT + Amn(n+1)p\left(1 + \frac{ap}{2}\right)\frac{dT}{T^{n+1}} - \frac{Am(n+1)}{T^n}(1 + ap)dp - Acdp$$

The integral of this exact differential is readily found to be

$$i = aT + \frac{1}{2}\beta T^2 - \frac{Am(n+1)}{T^n} p\left(1 + \frac{a}{2}p\right) - Acp + i_o \qquad [19]$$

29 The equation for intrinsic energy is derived from the relation

$$Au = i - Apv$$

Substituting for i the expression given by [10], we obtain for the energy expressed in B.t.u.

$$u = T\left(a + \frac{1}{2}\beta T - AB\right) - \frac{Amp}{T^n}\left[n + (n-1)\frac{a}{2}p\right] + i_o \qquad [20]$$

30 For the entropy an expression is readily found from [17]. Thus

$$ds = \frac{dq}{T} = c_p \frac{dT}{T} - A\frac{\partial v}{\partial T} dp$$

Introducing in this equation the expressions previously derived for c_p and $\frac{\partial v}{\partial T}$, the result is

$$ds = \left(\frac{a}{T} + \beta\right) dT + Amn(n+1)p\left(1 + \frac{a}{2}p\right)\frac{dT}{T^{n+2}} - AB\frac{dp}{p} - \frac{Amn}{T^{n+1}}(1+ap)dp$$

The integral is

$$s = a \log_e T + \beta T - AB \log_e p - Anp\left(1 + \frac{a}{2}p\right)\frac{m}{T^{n+1}} + s_o \ldots\ldots\ldots [21]$$

31 The constants of integration i_o and s_o may be obtained by passing to the saturation limit. Thus let [10] be written

$$i = i' + i_o$$

where i' stands for the variable part of the second member. The constants a, β, m, n, and a being known, i' can be found for any assumed pressure and temperature. Let i' be calculated for a given temperature T_1 and the corresponding *saturation* pressure p_1; from the properties of saturated steam i is known for this same state, and i_o is at once determined by subtraction. In the same way the constant s_o may be determined.

APPENDIX No. 5

THE SATURATION VALUES OF i FROM FORMULA [10] COMPARED WITH THE VALUES GIVEN BY MARKS AND DAVIS

32 For an intermediate range of temperature 200 deg. to 300 deg. (see Table 1) the agreement between the two sets of values is all that could be desired. Even for the range 100 deg. to 400 deg. the discrepancy only slightly exceeds $\frac{1}{3}$ of one per cent. Below 100 deg. and above 400 deg., however, the discrepancy becomes relatively large. It is possible to distribute the difference between the ends and the middle of the temperature range; thus if i_o be taken as 884.2 formula [19] gives values at 32 deg. and 450 deg., about 0.35 per cent larger and at 212 deg. about 0.3 per cent smaller than the corresponding values of Marks and Davis. If these latter values are accepted as final, the agreement of the two sets of values for the range 32 deg. to 450 deg. with a maximum deviation of $\frac{1}{3}$ of one per cent might be taken as evidence of the validity of equation [10]. However, before passing final judgment it will be instructive to examine somewhat in detail the experimental data relative to the heat content of superheated steam.

33 In Fig. 4, which is substantially a reproduction of part of Fig. 5 in the paper of Dr. Davis, the curve represents the values of i calculated from formula [10], with $i_o = 887.3$. For convenience the C scale of temperature is used, and the range is from 60 deg. to 190 deg. cent. The ordinates are the values of $i - i_{100}$ calories and the points are located from the data contained in Tables 1 and 2 of the paper in question. The points indicated by the symbol * represent the measurements of Henning; the other points are those deduced by Dr. Davis from the throttling experiments of Grindley, Griessmann and Peake. In Fig. 5 the curve shows values of i in B.t.u. calculated from equation [10] and the points represent values of i deduced from experiments on the latent heat r by Dieterici, Griffiths, Smith and Henning. The range 32 to 212 deg. fahr. is covered.

34 Directing attention to Fig. 4, it is seen that the points representing Henning's experiments accord well with the points deduced from the throttling experiments with the exception of the points at 140 deg. and 180 deg. Dr. Davis asserts that the point at 180 deg. is not entitled to nearly so much weight as the others because of a defect in the experimental apparatus. Probably this assertion is well founded. On the other hand, the points deduced from the throttling experiments in the region of 180 deg. are perhaps slightly low. In deriving the values from which these points are located, Dr. Davis used the specific heat measurements of Knoblauch and Jakob. Near the saturation limit these values of c_p are undoubtedly too high at the higher pressures. The result of using lower values of c_p is to raise the points in question. The change, however, is slight and is not sufficient to bring the points upon the curve.

35 An examination of the derivative $\frac{di}{dt}$ along the saturation curve may afford another basis of comparison. From the second degree equation of Dr. Davis, namely,

$$i = 1150.4 + 0.3745(t-212) + 0.00055(t-212)^2$$

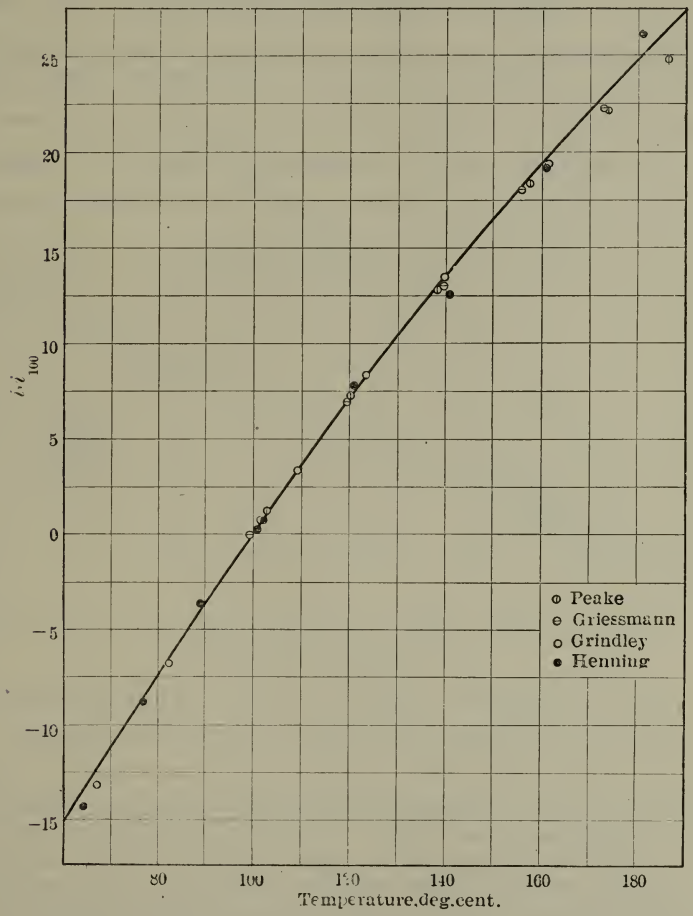


FIG. 4 CURVE REPRESENTING VALUES OF $i-i_{100}$ CALCULATED FROM [10]. THE POINTS GIVE VALUES DETERMINED FROM THE EXPERIMENTS OF HENNING, GRINDLEY, GRIESSMANN AND PEAKE

we have

$$\frac{di}{dt} = 0.3745 - 0.0011(t-212)$$

This linear relation is shown by the straight line A, Fig. 6. Writing [19] in the form $i = (p, T)$, we obtain

$$\left[\frac{di}{dt}\right]_{sat} = \left[\frac{di}{dT}\right]_p + \left[\frac{\partial i}{\partial p}\right]_T \left[\frac{dp}{dt}\right]_{sat} = c_p + \left[\frac{\partial i}{\partial p}\right]_T \frac{dp}{dt} \quad \ldots\ldots\ldots\ldots [22]$$

From [19] an expression for the derivative $\left[\frac{\partial i}{\partial p}\right]_T$ is readily found, and the final result is

$$\left[\frac{di}{dt}\right]_{sat} = c_p - \frac{dp}{dt}\left[\frac{Am(n+1)}{T^n}(1+ap)+c\right] \quad \text{.....[23]}$$

The values of this derivative for different temperatures ranging from 200 deg. to 425 deg. fahr. are shown by curve B, Fig. 6. The horizontal straight line C represents the Regnault value $\frac{di}{dt}=0.305$. The form of [23] is such that the relation between $\frac{di}{dt}$ and t cannot be represented by a straight line. Repeated attempts were made to change the constants α, β, m, n and a in such a way as to

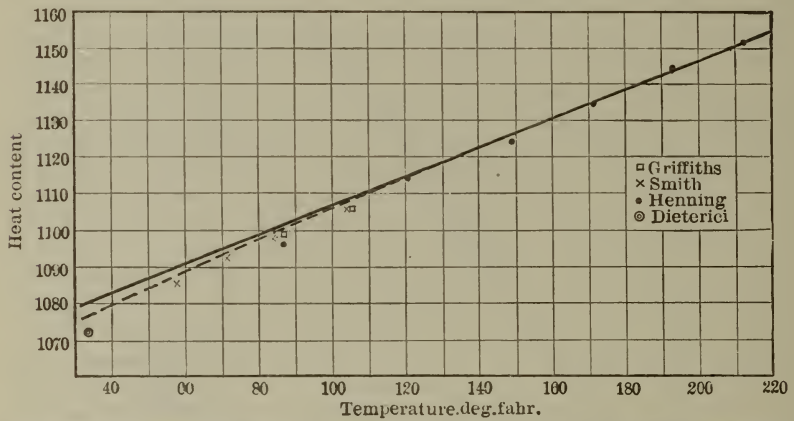


FIG. 5 CURVE REPRESENTING VALUES OF i IN B.T.U. CALCULATED FROM [10]. THE POINTS REPRESENT THE RESULTS OF VARIOUS EXPERIMENTS

straighten curve B and at the same time preserve the integrity of equations [3] and [7]. Such attempts were futile. The result of changing one constant would be neutralized by the necessary change in another, and the position of curve B would remain substantially unchanged. It follows that this curve must be accepted along with the characteristic equation [3] and the specific heat equation [7], or all three must be rejected. Curve B between the temperature limits shown may be approximately represented by a second degree equation though a third degree equation gives a closer agreement. Hence an approximate algebraic equation for i of the form $i=f(t)$ must be of the third degree at least.

36 The formula of Dr. Davis has been so universally accepted that a suggestion that it should be modified may seem presumptuous. The three points brought out in Pars. 33 to 35, however, lead to these conclusions: (*a*) The evidence furnished by Henning's point at 180 deg., whatever value may be attached to it, indicates that the Davis curve should be raised near 180 deg. (*b*) The newer and probably more accurate values of c_p near saturation if used

in determining the points deduced from the throttling experiments would raise these points near 180 deg. (*c*) Any equation of the form $i=f(p, T)$ deduced from a characteristic equation and an equation $c_p=\phi(p, T)$, as [10] was deduced, cannot have a derivative $\left[\frac{di}{dt}\right]_{sat}$ which is a linear function of the temperature. It is perhaps too much to assert that equation [10] is more accurate than the

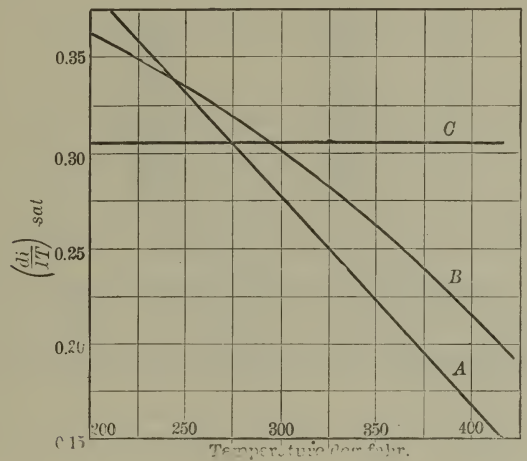


FIG. 6 CURVES FOR $\left(\frac{di}{dT}\right)_{sat}=f(t)$

A from the Davis Equation; *B* from [19]; *C* from Regnault's Equation

Davis formula for temperatures above 212 deg., but it is clear that the discrepancy between the values of i in Table 1, for the higher temperature, may be due in part to errors in the Marks and Davis values of i; it therefore furnishes no adequate basis for a rejection of the analysis leading to equation [10].

37 For temperatures below 212 deg. it is seen by reference to Fig. 5 that the curve derived from [19] fits very closely Henning's points from 120 deg. to 212 deg. Below 120 deg. the curve lies somewhat above the points determined by Dieterici, Griffiths, and Smith. It is probable, therefore, that for temperatures below 120 deg. equations [7] and [10] require modification. It may be pointed out that throughout the entire range, equation [10] represents the experiments of Henning about as well as any single empirical formula that could be produced.

38 Equation [21] gives the entropy of superheated steam for assumed values of p and T and may therefore be used to obtain the entropy of saturated steam. It will be of interest to compare the values thus determined with the values obtained in the usual way from the properties of saturated steam. This comparison is shown in Table 4. In the second column are given values from the Marks and Davis table; in the third column values obtained by using in the term

TABLE 4 ENTROPY OF SATURATED STEAM

Temperature, Deg. Fahr.	Entropy at Saturation		
	Marks and Davis	Marks and Davis with Modified Values of r	Calculated from [21]
100	1.9800	1.9869	1.9856
150	1.8674	1.8706	1.8706
200	1.7761	1.7765	1.7766
212	1.7565	1.7565	1.7565
250	1.6996	1.6992	1.6990
300	1.6343	1.6346	1.6342
350	1.5780	1.5800	1.5793
400	1.5286	1.5327	1.5322
450	1.4850	1.4911	1 4911

$\frac{r}{T}$, values of r derived from [10], and in the fourth column values calculated directly from [21]. The constant s_o is found to be -0.3960. The agreement between the corresponding values in the last two columns scarcely requires comment.

APPENDIX No. 6

VALIDITY OF EQUATION [7] AND VALUES OF CONSTANTS α AND β; ALSO FURTHER TESTS OF THE EQUATION FOR SPECIFIC HEAT

39 The evidence furnished by the Knoblauch and Mollier experiments is clearly in favor of the validity of formula [7] and justifies the values chosen for the constants α and β. The slope of the c_p curves at high temperature is evidently much smaller than was assumed in the original empirical curves of Knoblauch and Jakob and, for the temperature range under consideration, appears to be given correctly by the assumed value of β, namely $\beta = 0.00018$. There is still to be accounted for a considerable discrepancy between the slope thus determined and the slope of the Holborn and Henning curve. Holborn and Henning really determined, not the specific heat of steam directly, but the ratio of this specific heat to the specific heat of air under the same conditions. Comparing the results of Holborn and Henning on the specific heat of air with the results obtained by Joly and Swann, Callendar [1] makes the following statement: "The value found for the mean specific heat of air over the range 115 deg. to 270 deg. cent. by Holborn and Henning was 0.2315, which is about 5 per cent smaller than the probable value over this range. . . . . If the method gives a probable error of 5 per cent over the range 115 deg. to 270 deg., it does not seem at all impossible that the error may amount to 10 per cent over the range 115 deg. to 1400 deg. cent." Callendar's argument seems reasonable, and it is a fair conclusion that the experiments of Knoblauch and Mollier should be accepted as more worthy of confidence than the experiments of Holborn and Henning.

40 Formula [7] may be subjected to several other tests, namely: (*a*) The saturation values of c_p may be checked by Planck's thermodynamic relation; (*b*) the spacing of the c_p curves for different pressures may be tested by a method which connects c_p with the Joule-Thomson coefficient μ. This method is due to Dr. Davis.[2] (*c*) A relation between c_p, μ, and the heat content i of saturated steam affords a check on the values of c_p near the saturation limit at very low temperatures.

41 The values of c_p at saturation are given by Planck's equation, viz:

$$(c_p)_{sat} = \frac{dH}{dt} - \frac{r}{T} + AT\frac{dp}{dt}\left(\frac{\partial v}{\partial T}\right)_p \quad \ldots\ldots\ldots\ldots\ldots\ldots [24]$$

This equation was used effectively by Dr. Davis in showing that Knoblauch's values are preferable to those given by Thomas. It will be observed that with

[1] Appendix to the Report of the British Association Committee on the Investigation of Gaseous Explosions, Clerk's Gas, Petrol, and Oil Engines, pp. 361–367

[2] Proc. Amer. Acad., vol. 45, p. 290.

the exception of the derivative $\left[\frac{\partial v}{\partial T}\right]_p$, which may be obtained from [3], the terms in [22] are obtained from the properties of saturated steam. The total heat H and heat content i_{sat} of saturated steam are connected by the relation

$$H = i_{sat} - Apv'$$

in which v' denotes the specific volume of water; hence the derivative $\frac{dH}{dT}$ is found from the relation

$$\frac{dH}{dT} = \left[\frac{di}{dt}\right]_{sat} - Av'\frac{dp}{dt} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [25]$$

In the calculation of the values in Table 5 two courses were followed: (*a*) the derivative $\left[\frac{di}{dt}\right]_{sat}$ was found from the Davis equation

$$i = 1150.4 + 0.3745(T-212) - 0.00055(T-212)^2$$

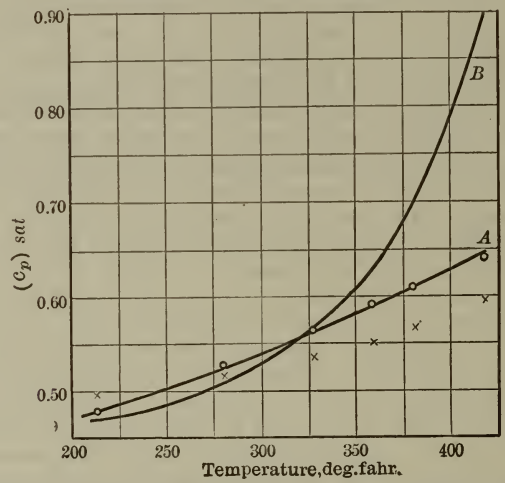


FIG. 7 CURVES SHOWING VARIATION OF $(c_p)_{sat}$ WITH THE TEMPERATURE

and values of $\frac{r}{T}$ were taken directly from the Marks and Davis tables; (*b*) the derivative was found from [10] (see [23], Appendix No. 6) and values of the latent heat r were obtained by subtracting from i_{sat} calculated from [10] the heat content of the liquid given in the Marks and Davis table. Table 5 shows the results of the calculation.

42 In Fig. 7, curve *A* represents the values of $(c_p)_{sat}$ calculated from [7] and curve *B* the values given by Knoblauch and Jakob. The points near curve *A* represent the values in column 3 of Table 5 and the points indicated by crosses represent the values in column 2. The agreement between curve *A* and the first set of points is sufficiently close to afford a confirmation of equation [7] at the saturation limit. For temperatures above 350 deg. curve *B* is clearly inadmissible. While the values of $(c_p)_{sat}$ obtained from Planck's equation [24] are in doubt, depending as they do on values of i and r that are not precisely known

it is impossible to bring the points calculated from [24] in harmony with curve *B*. For temperature above 212 deg., at least, equation [7] satisfies the test furnished by Planck's relation.

43 The specific heat curves derived from equation [7] may be tested by a method developed by Dr. Davis in the paper heretofore cited. It is based upon the following relation between c_p and the Joule-Thompson coefficient

$$c_p = (c_p)_o e^{-\int_{p_o}^{p} \left(\frac{\partial \mu}{\partial t}\right)_p dp} \quad [26]$$

By means of this relation the ratio $\frac{c_p}{(c_p)_o}$ may be calculated for assumed values

TABLE 5 VALUES OF c_p AT SATURATION

Pressure, Lb. per Sq. In.	Calculated from [22]		Calculated from [7]	Knoblauch and Jakob Curves
	Using Values from Marks and Davis	Using Values from [19]		
15	0.4970	0.4796	0.4775	0.472
50	0.5071	0.5241	0.5228	0.508
100	0.5343	0.5661	0.5604	0.565
150	0.5503	0.5903	0.5876	0.630
200	0.5677	0.6106	0.6097	0.705
300	0.5973	0.6410	0.6454	0.898

of p and T along a curve of constant heat content i. The specific heat $(c_p)_o$ is taken at an assumed standard pressure, which was assumed by Dr. Davis as 1 kg. per sq. cm. Evidently the method gives only relative results; that is, it does not give the absolute position of any c_p curve, but rather the spacing of the family of c_p curves. Using the data given by Dr. Davis in Table 5 of his paper, the various ratios $\frac{c_p}{(c_p)_o}$ were calculated from equation [7] and the results obtained were compared with the results derived from the calculations of Dr. Davis. Table 6 shows the two sets of values.

An inspection of Table 6 shows close agreement at the lower pressures and some divergence at the higher pressures. This test, while of value in a general way, must be taken with some reservation. In the first place the value of the coefficient μ is not known with absolute accuracy. Secondly the constant i curve that must be laid down before the calculation is started depends upon the assumed saturation values of i. If i_{sat} is determined from [10] instead of from the Davis formula, the constant i curves will lie nearer the saturation curve and as a result the values of $\frac{c_p}{(c_p)_o}$ calculated from equation [7] will be larger at the higher pressures.

44 A test that applies at low temperatures near the saturation limit is furnished by Griessman's equation

$$c_p = \frac{\frac{\partial i}{\partial T}}{1 - \mu \frac{dp}{dT}}$$

in which μ denotes the Joule-Thomson coefficient. This relation is in reality identical with that given by equation [22] and may be derived from [22] with the aid of the defining relation $\mu = \left[\frac{\partial T}{\partial p}\right]_i$. Since μ and c_p are intrinsically positive, Griessman's equation imposes the condition that at the saturation limit c_p must be numerically greater than the derivative $\left[\frac{di}{dT}\right]_{sat}$. On page 290 of his paper Dr. Davis concludes from this condition that at 32 deg. fahr. $(c_p)_{sat}$ must be as great as 0.44. From [7] the values of $(c_p)_{sat}$ is about 0.417. A consideration of

TABLE 6 COMPARISON OF c_p RATIOS

Pressure, Kg. per Sq. Cm.	Curve 1			Curve 2			Curve 3			Curve 4		
	Temperature, Deg. Cent.	$\frac{c_p}{(c_p)_0}$		Temperature, Deg. Cent.	$\frac{c_p}{(c_p)_0}$		Temperature, Deg. Cent.	$\frac{c_p}{(c_p)_0}$		Temperature, Deg. Cent.	$\frac{c_p}{(c_p)_0}$	
		Davis	From [7]		Davis	From [7]		Davis	From [7]		Davis	From [7]
0.1	121.3	0.946	0.945	149.0	0.960	0.962	204.5	0.984	0.981	287.8	0.994	0.993
0.5	121.3	0 970	0.970	150.5	0.979	0.979	205.4	0.991	0.990			
1.0	125.8	1.000	1.000	152.3	1.000	1.000	206.4	1.000	1.000	288.9	1.000	1.000
2.0	130.6	1.060	1.053	155.8	1.041	1.035	208.5	1.018	1.020	290.1	1.007	1.006
3.0	135.1	1.120	1.10	159.1	1.082	1.081	210.5	1.034	1.039			
3.5	137.2	1.150	1.122									
6.0				162.3	1.122	1.107	212.5	1.051	1.057	292.4	1.020	1.023
6.0				168.5	1.200	1.165	216.5	1.084	1.091	294.7	1.032	1.038
8.0				174.3	1.264	1.215	220.2	1.118	1.123	297.0	1.044	1.052
10.0				179.7	1.344	1.260	223.9	1.151	1.151	299.2	1.057	1.066

this discrepancy throws light on the question heretofore raised relative to heat content at the lower temperatures. The minimum value 0.44 assigned by Mr. Davis is evidently derived from the value of the derivative $\left[\frac{di}{dT}\right]_{sat}$ at 32 deg.; that is, it is the slope of the curve giving the saturation values of i (see Fig. 5) at 32 deg. As the values of i near 32 deg. are very uncertain, no great weight can be attached to this particular number 0.44. From the connection between the equations [7] and [10], Griessman's relation is automatically satisfied at all temperatures by values of c_p calculated from [7] and values of i calculated from [10]; hence at 32 deg. the slope of the curve $i_{sat} = f(T)$, Fig. 5, must be less than 0.417. If, however, the curve were lowered to agree more closely with the experimental points the slope at 32 deg. would be increased, and as a result c_p would be raised. By an arbitrary change in the function $\phi(T)$ within the range of 32 deg. to 212 deg. the curve of Fig. 5 may be wrenched into closer agreement with the points and $(c_p)_{sat}$ at 32 deg. may be correspondingly raised. In [7]

ψ (T) was taken as $0.367+0.0001T$. Let this be replaced by the function $\psi(T) = 0.501-0.0001T$; then in [10] the first two terms for the expression for i become

$$\int \psi(T)dT = 0.501T - 0.00005T^2 + C$$

The straight lines representing $\phi(T)$ and $\psi(T)$ (see Fig. 8) intersect at $t=210$ deg. ($T=670$); hence the constant C is determined by the condition that the value of i for $T=210$ deg. shall be the same when calculated by the modified equation as when obtained from [10]. The effect of this modification of formula [10] is shown by the dash curve, Fig. 5. The resulting value of $(c_p)_{sat}$ at 32 deg. is 0.452, which is very nearly the value taken by Marks and Davis.

45 The process just described is wholly artificial and cannot be justified on any rational basis. There is no reason to suppose that the function $\phi(T)$ is properly represented by a broken line, as shown in Fig. 8. It is conceivable that for the range of temperature under consideration the function should be represented by a curve somewhat of the form shown in the figure; but in the absence of accurate measurements of i or c_p in this region, it is not worth while to complicate the equations by assuming a less simple form for the function $\phi(T)$. The recent work of Smith [1] points to the conclusion that all measurements of the

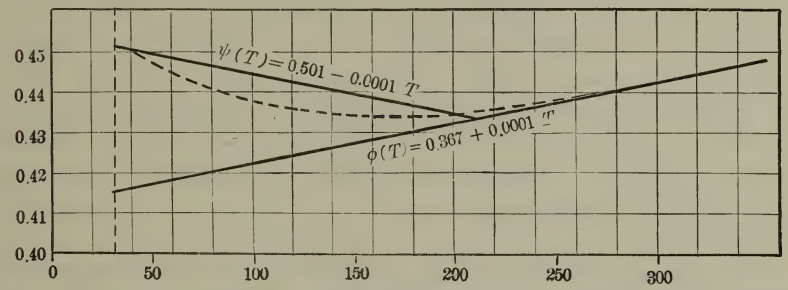


FIG. 8 CURVES REPRESENTING POSSIBLE FORMS OF THE FUNCTION ψ (T) NEAR 32 DEG. FAHR.

latent heat are probably somewhat in error and that all the points in Fig. 5 are too low; hence the exact value of i near 32 deg. are uncertain. No accurate measurement of $(c_p)_{sat}$ near 32 deg. is available. At present, therefore, we must be satisfied with the following general conclusions:

- *a* The value of $(c_p)_{sat}$ at 32 deg. probably lies between 0.42 and 0.45.
- *b* The curve representing $\phi(T)$ probably lies somewhat above the line $\phi(T)=0.367+0.0001T$ (Fig. 8).
- *c* The curve that represents the relation $i_{sat}=f(T)$ lies very close to the curve of Fig. 5. If Smith's criticism of the latent heat measurement is just, the curve as a whole should be raised.

46 The most important conclusion that may be derived from this discussion is that the analysis is adequate to furnish consistent results when accurate experimental data are available. With accurate measurements of i near 32 deg., the function $\phi(T)$ may be determined and from this accurate value of c_p may be obtained. Conversely, if c_p is accurately determined at 32 deg. and at points in that vicinity, so that the function $\phi(T)$ may be established, then the course of the curve for i_{sat} will be determined.

[1] Physical Review, September 1911.

DISCUSSION

H. N. Davis.[1] Professor Goodenough's paper is of great value, and the characteristic equation which he proposes is by far the best yet suggested for steam. Of previous characteristic equations, several, notably Zeuner's and Tumlirz's, have been of such a form as to yield simple expressions for the derivatives and a beautiful thermodynamic theory, but have not given a sufficiently accurate representation of the facts. On the other hand, any attempt to follow the facts more closely has usually led to a much less convenient form of equation. Professor Goodenough has been able to devise an equation that has both advantages. It represents the experimental volumes quite as well as any other, and it is particularly well adapted to thermodynamic calculations.

Professor Goodenough is moreover more in a position to attack a more fundamental problem than those who have proposed most previous characteristic equations, for they had only one sort of experimental data to work from, namely specific volumes. Even in Linde's time (1905) the true values of the total heat of saturated steam were less certain than at present, and Knoblauch's valuable measurements of c_p had not been published. Professor Goodenough has formulated not merely one of these sorts of experimental data, but all three.

The devising of such a system of equations is second in importance only to the establishment of the true experimental values themselves, for not only are closed expressions for the various functions valuable to engineers, but by far the easiest way to compute a steam table is through such empirical equations whenever they are available. The writer has therefore been interested in verifying a considerable number of Professor Goodenough's computations, and has been much impressed by their thoroughness and accuracy and by the fairness of his discussion of the results.

It should, however, be remembered that no empirical equations can ever be more trustworthy than the data on which they are based. Professor Goodenough has succeeded remarkably well in adjusting his system of equations to all the three kinds of known data over a large range of pressure and temperature. But nevertheless, in the regions where his system fails to represent trustworthy facts, one should not be misled into assigning a greater validity to the equations than to the facts. It is therefore important to consider with some

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care the results of the various tests to which he has subjected his proposed formulae.

In the first place, there is no question but that his characteristic equation represents the volume measurements within their limit of error. This is not, however, an assurance that the equations deducible thermodynamically from his characteristic equation will repre-

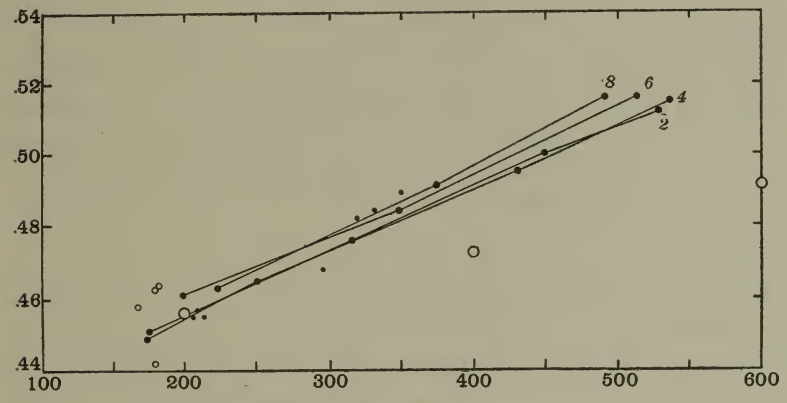

FIG. 9 c_p AT ZERO PRESSURE

THE VALUES OF c_p AT ZERO PRESSURE, AS REDUCED FROM THE OBSERVED VALUES AT HIGHER PRESSURES BY MEANS OF PROFESSOR GOODENOUGH'S EQUATION, ARE PLOTTED AGAINST THE CENTIGRADE TEMPERATURE. THE LARGER BLACK DOTS ARE THE OBSERVATIONS OF KNOBLAUCH AND MOLLIER, REDUCED FROM THE PRESSURES INDICATED, IN KG. PER SQ. CM. THE SMALLER BLACK DOTS ARE THE OLDER OBSERVATIONS OF KNOBLAUCH AND JAKOB. IN BOTH CASES ONLY MEANS OF GROUPS OF OBSERVATIONS OF SIMILAR PRESSURE AND TEMPERATURE ARE PLOTTED. THE THREE LARGER CIRCLES ARE HOLBORN AND HENNING'S VALUES; THE FOUR SMALLER CIRCLES NEAR 175 DEG. ARE REGNAULT'S OBSERVATIONS RECOMPUTED WITH MODERN VALUES FOR THE SPECIFIC HEAT OF WATER.

sent adequately the specific and total heats. Linde's equation also represents the volume measurements within their limit of error, and almost certainly gives erroneous values for c_p. And a form of Tumlirz's equation, proposed by Linde, represents the volumes almost as well, and yet demands that c_p shall not vary with pressure at all, which is certainly false. Professor Goodenough's specific and total heat formulae must therefore be considered on their own merits.

As to specific heats, two questions are involved: (*a*) Does his characteristic equation predict the right variation of c_p with pressure? (*b*) Has he properly evaluated the arbitrary function $\psi(T)$ in his equation [6] that comes in as a constant of integration and represents the variation of c_p with temperature at zero pressure?

The easiest way to answer the first of these questions is to reduce

all the known measurements of c_p to zero pressure by means of Professor Goodenough's characteristic equation, and to plot the results against T. This has been done in Fig. 9. All the points should lie on a single smooth curve. It appears that Holborn and Henning's points are probably in error as Professor Goodenough says. Knoblauch's points do fall nearly together, much more so than would have been the case if Linde's equation had been used in the reduction, but nevertheless the high pressure points show an unmistakable tendency to run high. Therefore Professor Goodenough's equation is much better than Linde's, but is not perfect. It is undoubtedly sufficiently accurate for most computations involved in engineering problems. The inaccuracies are, however, such as may be expected to lead to erroneous values for the total heat of saturated steam at pressures above 4 kg. per sq. cm. (57 lb. per sq. in.) and temperatures above 143 deg. cent.

As to the second question, the points in the figure can be represented by a straight line about as well as by any other curve, and the line proposed by Professor Goodenough is as good as any other. But it should be remembered that all the points in the figure lie between 150 deg. and 500 deg. cent. In particular, absolutely nothing is known about the shape of the curve below 100 deg. cent., and many simple functions can be invented whose graphs are nearly straight between 150 deg. and 500 deg. and yet show considerable curvature below 100 deg. The broken line in Fig. 8 is by no means a fair sample of what could be done in this region, but even this broken line is quite as reasonable as the extrapolation of the original straight line so far below the limits of the data on which it is based. It is furthermore certain that the straight line can not be extrapolated an equal distance in the opposite direction. We know little enough about c_p at very high temperatures, but all we do know shows that it is not a linear function of the temperature. Therefore Professor Goodenough's straight line must be regarded as only a first approximation to the truth. In the region of importance to engineers it is good enough, but at low and at high temperatures it is probably wrong.

Finally, as to total heats, at temperatures between 80 deg. and 160 deg. cent. the agreement of Professor Goodenough's values for i_{sat} with the facts is remarkably good. Above 170 deg. cent. they run higher than the values which the writer has previously suggested. This, however, is exactly the region where the proposed formulation of c_p has just been shown to be somewhat in error

and where erroneous values for i_{sat} were to be expected. It is therefore probable that Professor Goodenough's values of i_{sat} are less reliable than those previously suggested. At still higher temperatures it is certain that his formula for the total heats fails to represent the facts, for at the critical point ($T_c = 1160$ deg. fahr., $P_c = 3188$ lb. per sq. in.), according to his formula i is about 1170 B.t.u. whereas it is known to be about 900 B.t.u.[1] His values of i_{sat} must therefore run increasingly too high at very high temperatures, and it is by no means unreasonable that some error of this sort is already apparent at 190 deg. cent. It is probable that this imperfection above 160 deg. cent. cannot be removed by any reasonable alteration in the function $\psi(T)$, but is inherent in the characteristic equation itself.

Below 80 deg. cent., also, his values for i_{sat} run high, until at 0 deg. cent. the discrepancy is nearly 1 per cent. In this region we have seen that his assumed values of $\psi(T)$ and therefore of the total heat at saturation have no experimental foundation whatever. Neither, for that matter, has the characteristic equation itself any experimental foundation below 110 deg. cent. There can be no question but that the previously accepted values of the total heat should stand unchanged. Even the objection which he urges on page 491 that all measurements of total heats are uncertain because of priming, is less valid at low temperatures than at or above 100 deg. cent. because of the small density of the steam, and, in the case of Smith's measurements at least, because of the experimental methods used. Therefore the argument quoted by Professor Goodenough on page 490 is still valid, and $(c_p)_{sat}$ at 0 deg. cent. should probably be as high as 0.44. It is very possible that this imperfection below 80 deg. cent. could be removed by a suitable change in $\psi(T)$, as he suggests.

Incidentally Professor Goodenough is quite right in his statement that the derivative $\left(\frac{di}{dT}\right)$ is not a linear function of the temperature (curve A, Fig. 6), as a second degree formula for i_{sat} suggests, but should have a shape somewhat like his curve B. The writer's straight line A represents the data on which it was based just about as well as Professor Goodenough's straight line mentioned above represents the value of c_p at zero pressure, and no better. That is, the second degree equation is a good first approximation for i itself within a limited range, but cannot be safely differentiated, and this was all it was intended to do. The differentiation

[1] See either Davis, Proc. Am. Acad., vol. 45, 1910, p. 283, fig. 8, or Schüle, Z. d. V. d. I., 1911, p. 1512, fig. 4.

of an empirical formula is always hazardous, and must be justified on its own merits. A still more accurate formula, expressed in terms of the latent heat, which can be differentiated, was proposed in the paper before the American Academy referred to above.

To sum up, Professor Goodenough is the first who has ever succeeded in reconciling the accepted volume, specific heat, and total heat measurements through a thermodynamically self-consistent set of equations over any range whatever. He has done this over a very large range. Within this range his equations can be used with confidence, and are very valuable. This range does not, however, in the writer's opinion, extend much above the pressure corresponding to a saturation temperature of 160 deg. cent. (320 deg. fahr.), namely about 90 lb. per sq. in., or much below that corresponding to 80 deg. cent. (176 deg. fahr.), namely about 7 lb. per sq. in. Even up to 225 lb. per sq. in. (saturation temperature 200 deg. cent. or 392 deg. fahr.), the errors in his formulae are not serious, although for theoretical work, especially if it involves differentiations, the formulae should be used with caution. At pressures below 7 lb. per sq. in. and at all pressures when the superheat is greater than 400 deg. cent. (720 deg. fahr.) the formulae should be regarded only as approximations to the truth. The proposed changes in the accepted values of the total heat of saturated steam above 160 deg. cent., and below 80 deg. cent. do not seem to be justified.

R. C. H. Heck. Having worked over the whole ground covered by this paper to the extent of laying out a complete set of steam tables,[1] the writer has been much interested in making a critical examination of the conclusions here presented and in preparing graphical comparisons between his own results, the tables of Marks and Davis, and the formulae of Professor Goodenough.

The most serious fault to be found with the system of steam properties proposed by Goodenough is the apparent complete lack of an effort to satisfy Clapeyron's equation connecting latent heat r with saturation volume, namely

$$\frac{r}{u} = AT\frac{dp}{dt} \qquad [A]$$

where u is the volume increase from water to steam. By far the most precisely determined physical relation for steam is that between pressure p and temperature t of saturation; and the p-t formula

[1] Published in The Steam Engine and Turbine, 1911.

established by Marks [1] represents the recent very reliable and accurate data with such precision as to leave nothing better to be desired in the way of an exact determination of $\frac{dp}{dt}$.

The disaccord at saturation between volume and heat quantities

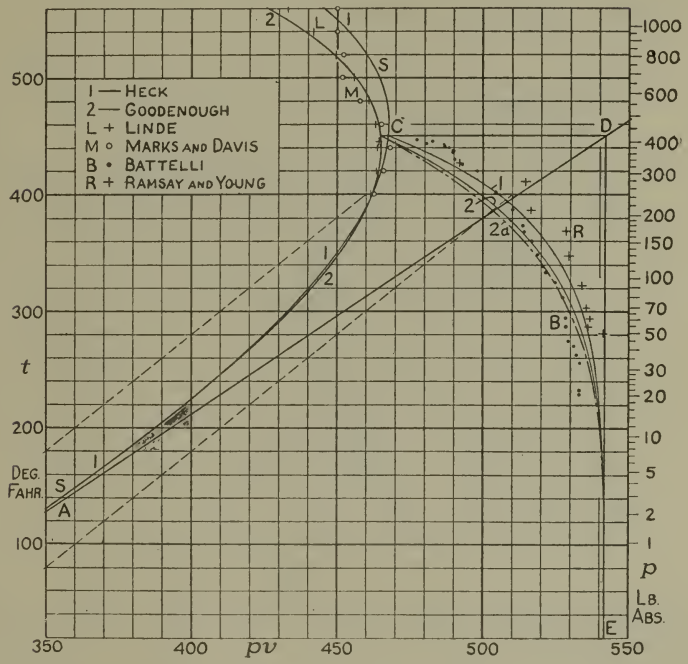

FIG. 10 PLOT OF pv ON t FOR 1 LB. OF STEAM, WITH p IN LB. PER SQ. IN. ABSOLUTE, v IN CU. FT., t IN DEG. FAHR. LINE AD REPRESENTS THE IDEAL RELATION $pv = BT$, CURVES SS ARE SATURATION LINES, CD IS AN ISOTHERMAL.

CURVES CE ARE ISOTHERMALS IN A PLOT OF pv ON p, BUT WITH p TO THE VARIANT SCALE AT THE RIGHT, WHICH SHOWS THE SATURATION PRESSURE CORRESPONDING TO TEMPERATURE AS SCALED UNIFORMLY AT THE LEFT. THE EXPERIMENTAL POINTS PLOTTED ALONG THESE CURVES SHOW ISOTHERMAL OBSERVATIONS OF VOLUME, BY BATTELLI AT 448.5 DEG. FAHR. AND BY RAMSAY AND YOUNG AT 446 DEG. FAHR.

calculated by the formulae on page 474 is made strikingly apparent in Fig. 12, where curve 2 represents formula [*c*] and curve 3 formula [*a*], as more fully described beneath the diagram. Over the range of best volumetric experiment, or from 360 deg. to 212 deg. and lower, there is, to be sure, agreement within 0.2 to 0.4 per cent of

[1] **The Journal, May 1911, p. 573.**

latent heat r. But this agreement seems to be fortuitous, because the physical data are of that degree of concord, rather than intentional. Above 360 deg. the two curves diverge excessively, with the blame about equally divisible between characteristic equation and formula for heat content: in the low range there is similar but smaller difference in trend, due to the unwarranted departure of the i_{sat} curve from the result of experiment as laid out in Fig. 13.

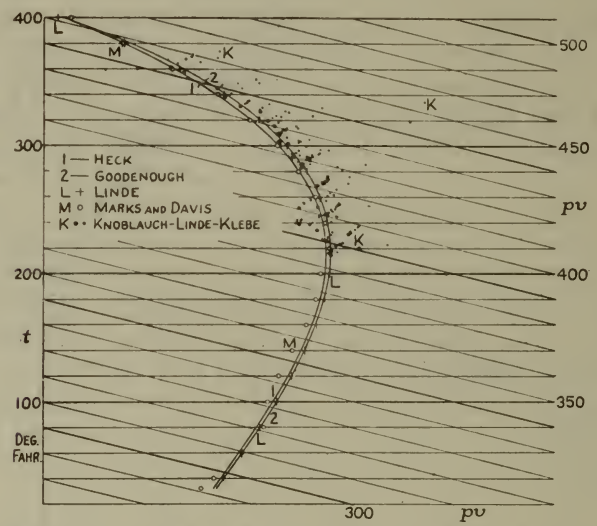

FIG. 11 THE PORTION OF FIG. 10 BOUNDED BY DOTTED OUTLINE, WITH INCREASED HORIZONTAL SCALE AND WITH LINES OF EQUAL pv INCLINED. PLOTTED K POINTS SHOW EXPERIMENTS OF KNOBLAUCH, LINDE AND KLEBE (AS IN FIG. 14 ALSO), WITH ONLY A SHORT RANGE NEAR SATURATION GIVEN IN MOST CASES.

IN FIGS. 10 AND 11, ONE HORIZONTAL DIVISION EQUALS FROM 3.3 PER CENT (AT BOTTOM) TO 2.2 PER CENT OF THE SATURATION VALUE OF pv.

THE CHARACTERISTIC EQUATION

The general idea governing the development of a pressure-volume-temperature equation for steam, most simply written in the form

$$pv = BT - f(p, T) \quad \ldots\ldots\ldots\ldots\ldots\ldots [B]$$

is to give the constant B the value which it would have for a perfect gas, according to the molecular weight of H_2O, and then make the corrective term $f(p, T)$ represent and measure the departure of actual from ideal substance. The rational requirement that steam shall approach the law $pv = BT$ as it gets farther from saturation and as density becomes less calls for a decrease of $f(p, T)$ with increase of T and with decrease of p.

The Munich volumetric experiments, upon which are based Linde's and Goodenough's characteristic equations, are represented in Fig. 14 by a direct plot of pressure and temperature. The shortness of the range from saturation, out into the region of superheat,

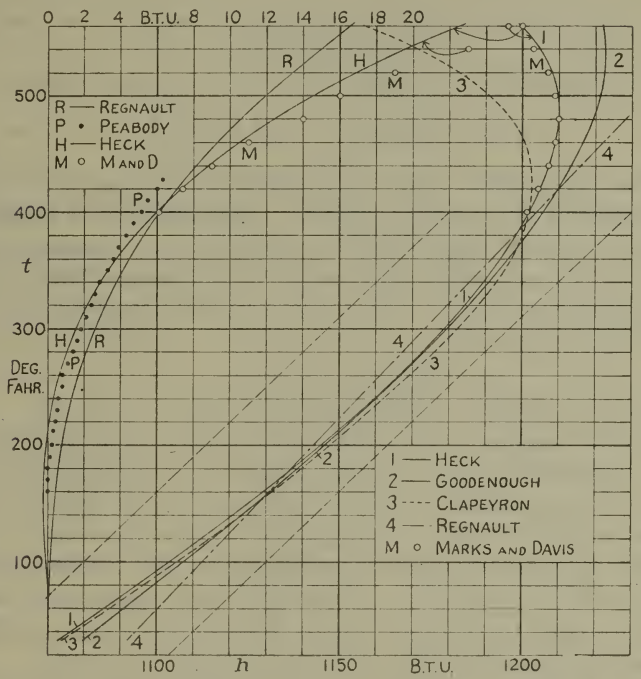

FIG. 12 MAIN GROUP OF CURVES, NOS. 1 TO 4, HEAT CONTENT AT SATURATION (SYMBOL h INSTEAD OF i) LAID OUT ON TEMPERATURE t:

CURVE 1, BY WRITER, IN CLOSE ACCORD WITH MARKS AND DAVIS.

CURVE 2, FROM GOODENOUGH'S FORMULA [c] FOR HEAT CONTENT, EVALUATED AT SATURATION.

CURVE 3, FROM VOLUMES AS GIVEN BY GOODENOUGH'S FORMULA [a], BY MEANS OF CLAPEYRON'S EQUATION, USING p AND $\frac{dp}{dt}$ FROM MARKS' EQUATION IN THE JOURNAL FOR MAY 1911.

CURVE 4, THE OLD STRAIGHT-LINE FORMULA OF REGNAULT.

SECOND GROUP OF CURVES, AMOUNT BY WHICH THE HEAT OF LIQUID q EXCEEDS THE TEMPERATURE-RISE $(t-32)$ TO B.T.U. SCALE AT TOP:

CURVE R, REGNAULT'S FORMULA.

CURVE P, PEABODY, BASED ON REGNAULT'S DATA.

CURVE H, BY WRITER, SAME DATA AS MARKS AND DAVIS, ESSENTIAL AGREEMENT EXCEPT ABOVE 440 DEG.

is at once apparent. The most that these constant-volume point series can do is to establish the initial trend of curves which in the distance shall become asymptotes to straight lines from the ideal equation $pv = BT$. Since at saturation we have the double resurcee

of direct measurement of volume and of indirect determination through Clapeyron's relation, the idea of the saturation line as furnishing well located starting points for any superheat curves is to be strongly emphasized.

In Linde's equation, the term $f(p, T)$ is separable into two distinct factors which are respectively functions of p and T, or we may write

$$f(p, T) = F_p \times F_t \quad \ldots\ldots\ldots\ldots\ldots\ldots [C]$$

The writer has retained Linde's value of the pressure factor

$$F_p(p + 0.0014\ p^2) \text{ or } p\left(1 + \frac{p}{700}\right) \quad \ldots\ldots\ldots\ldots\ldots [D]$$

then after adjustment of the pv_{sat} curve the difference $(BT - pv_{sat})$ was divided by F_p in order to get F_t. This scheme was followed up to nearly 550 deg. fahr., with the results shown comparatively in Fig. 15: extension into the high range is nothing more than a graphical extrapolation of F_t. Since Goodenough's corrective term would take the general form

$$f(p, T) = ap + f_p f_t \quad \ldots\ldots\ldots\ldots\ldots\ldots [E]$$

it cannot be included in the comparison made in Fig. 15.

While the diagram of pv on t is a most effective instrument for the study of saturated steam, we must change to pv on p in order to get a useful representation of the isothermal, which in Fig. 10 at first appears as line CD. The obvious thing is a uniform scale of p with the curve in true form and showing exactly the influence of p in $f(p, T)$. But for purposes of illustration and comparison within a moderate space, the scheme used in the right-hand part of Fig. 10 and more extensively in Fig. 16 is rather more convenient.

In Fig. 10, curve 1 from C to E is essentially Linde's isothermal, determined by F_p as written in equation $[D]$; curve 2 is from Goodenough's equation, with the smaller curvature resulting from the use of 0.0006 instead of 0.0014 as coefficient of p^2; and line $2a$ would be a straight line if p were measured to a uniform scale. At low and moderate pressures the curvature of the isothermal with p is so small that it can vary considerably within the limit of observational irregularity of the Munich experiments and within their short range. But the Ramsay and Young results, as plotted in Fig. 16, seem to confirm the greater curvature of Linde rather than the less curvature of Goodenough. Certain it is, as the writer has found by trying out a tentative plot of pv on uniform p clear up to the critical pressure, that isothermals from 550 to 700 deg. will have to be curved inward quite markedly from Linde's form, at high pressures, in order to make them touch the probable saturation curve at their character-

istic temperatures: apparently, a pressure factor containing higher powers than p^2 is indicated.

On the whole, the conclusion seems justified that Goodenough's decrease in the curvature of the pv-p isothermals is a step in the wrong direction.

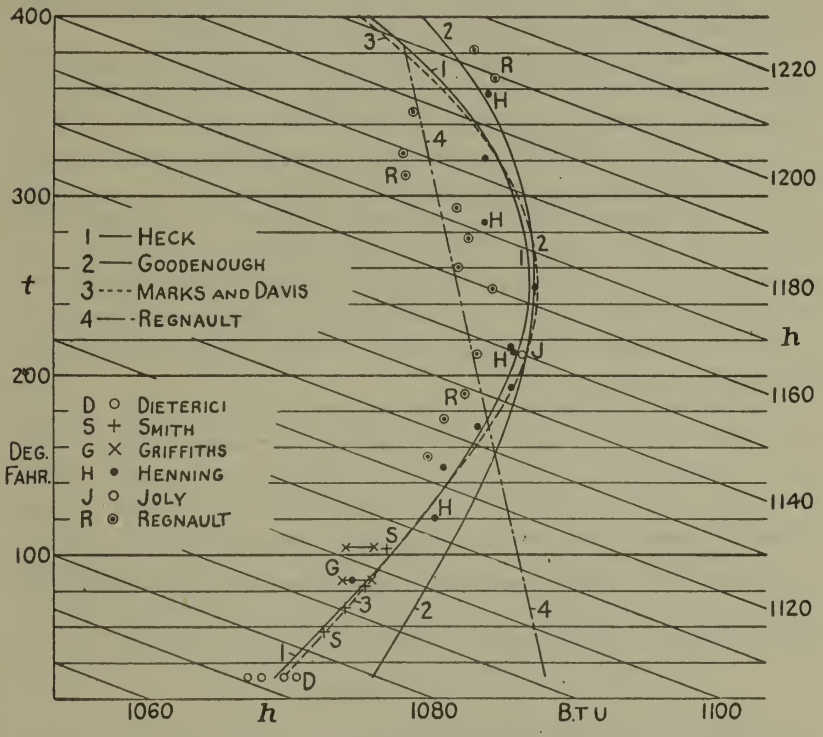

FIG. 13 PORTION OF FIG. 12 WITHIN DOTTED OUTLINE, HORIZONTAL SCALE ENLARGED AND LINES OF EQUAL h INCLINED. SAME CURVES AS IN FIG. 12, EXCEPT THAT NO. 3 NOW REPRESENTS THE TABLES OF MARKS AND DAVIS INSTEAD OF RESULTS FROM GOODENOUGH THROUGH CLAPEYRON. AS TO THE EXPERIMENTAL POINTS, THE FOUR AT 32 DEG. SHOW DIFFERENT INTERPRETATIONS OF DIETERICI'S MEAN RESULT, ACCORDING TO VALUE OF UNDERLYING PHYSICAL UNIT OF ENERGY; AND THE HENNING POINTS SHOW ORIGINAL VALUES, NOT INCREASED BY THE 1.3 OR 1.4 B.T.U. SUGGESTED BY SMITH.

IN FIGS. 12 AND 13, ONE HORIZONTAL DIVISION OF 10 B.T.U. EQUALS FROM 0.9 PER CENT, AT BOTTOM, TO 1.5 PER CENT OF LATENT HEAT r.

TOTAL HEAT AT SATURATION

There is no doubt in the writer's mind that Davis' formula for total heat from 212 deg. to 400 deg. is essentially correct, within the limit of 1 B.t.u. which he claimed for it. But with rigid adherence to that second-degree equation it is not possible to extend a smoothly

continuous curve beyond the limits named. As a means of adjustment to better form, the rate of change $\frac{dH}{dt}$ was therefore made to vary according to a smooth curve. It may be appropriate to remark that "total heat" is here being used with exactly the same meaning as "heat content" in the paper. As to symbol, h stands for total heat in general, H for the particular value at saturation.

Concerning the plot of experimental points in Fig. 13, a few notes may be added to those beneath the diagram. The double G points show lower values as originally reported and higher values as corrected by Smith. The Henning values are original, since Henning in his second paper [1] says that Smith's proposed correction is a mistake. The Regnault points are group averages as arranged by Smith.[2] While total heat H is laid out in Figs. 12 and 13, both experiment (except by Regnault) and calculation through Clapeyron's equation give latent heat r, to which must be added the liquid heat q. In all cases this q was taken from a tabular column represented by curve H in Fig. 12.

The lack of agreement between curves 2 and 3 of Fig. 12 has received sufficient comment. The difference between curves 1 and 3 in the same figure, below 360 deg., is due chiefly to volume differences represented by the pv curves 1 and 2 of Figs. 10 and 11; but below 212 deg. a little of it comes from the fact that the writer's values of $\frac{dp}{dt}$, worked out before the publication of Marks' equation, are not quite identical with results from that equation.

TOTAL HEAT OF SUPERHEATED STEAM

The best calorimetric experiments, while essentially correct, are not precise enough to make possible an accurate layout of a system of c_p curves in a purely empirical way: this is clearly evident from a glance at Fig. 13 and is the reason why some such scheme as that proposed by Goodenough must be resorted to. The writer has devised another and simpler method, of which the description is reserved for independent presentation, which is concerned primarily with total heat and uses c_p rather as a check. Developed from the work of Davis on the Joule-Kelvin coefficient μ, this device determines the shape and location of isothermal lines in

[1] Annalen der Physik, vol. 29, p. 445.

[2] Monthly Weather Review, October 1907.

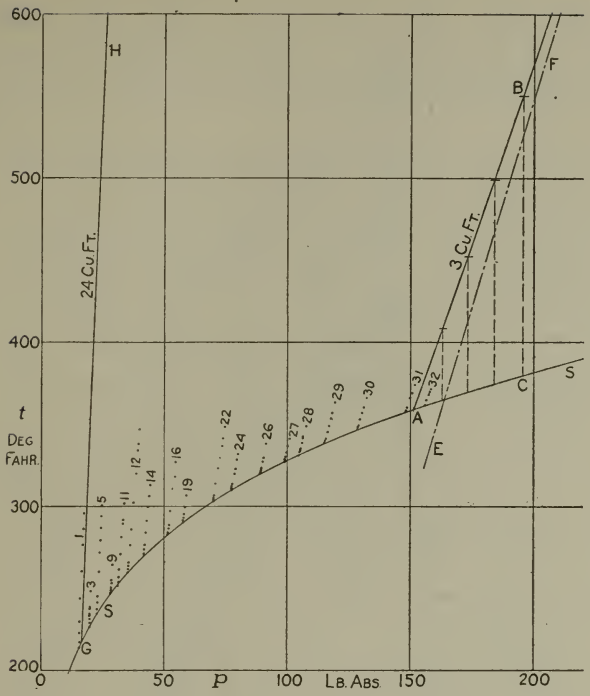

FIG. 14 PRESSURE-TEMPERATURE DIAGRAM FOR CONSTANT-VOLUME OBSERVATIONS, EXPERIMENTS OF KNOBLAUCH, LINDE, AND KLEBE AT MUNICH. BASE (VERTICAL) TEMPERATURE FAHRENHEIT, ORDINATE (HORIZONTAL) ABSOLUTE PRESSURE IN POUNDS PER SQUARE INCH ABSOLUTE.

CURVE *SS*, SATURATION LINE: *AB* AND *GH* SHOW THE $p-t$ RELATION FOR 1 LB. OF STEAM IN A VOLUME OF 3 CU. FT. AND OF 24 CU. FT. ACCORDING TO LINDE'S FORMULA: *EF* IS THE IDEAL LINE FOR 3 CU. FT., FROM THE EQUATION $pv=BT$. THIS *FE*, PRODUCED DOWNWARD, WOULD INTERSECT THE TEMPERATURE AXIS (OR LEFT-EDGE LINE OF ZERO PRESSURE) AT ABSOLUTE ZERO.

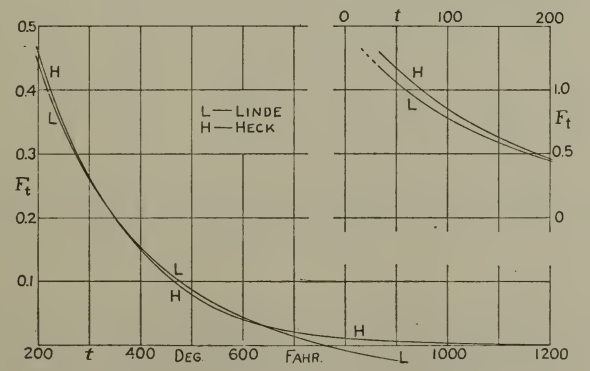

FIG. 15 TEMPERATURE FACTOR F_t IN LINDE'S CHARACTERISTIC EQUATION FOR STEAM. CURVE *L*, FROM THE ORIGINAL FORMULA, CURVE *H*, AS MODIFIED AND EXTENDED BY THE WRITER.

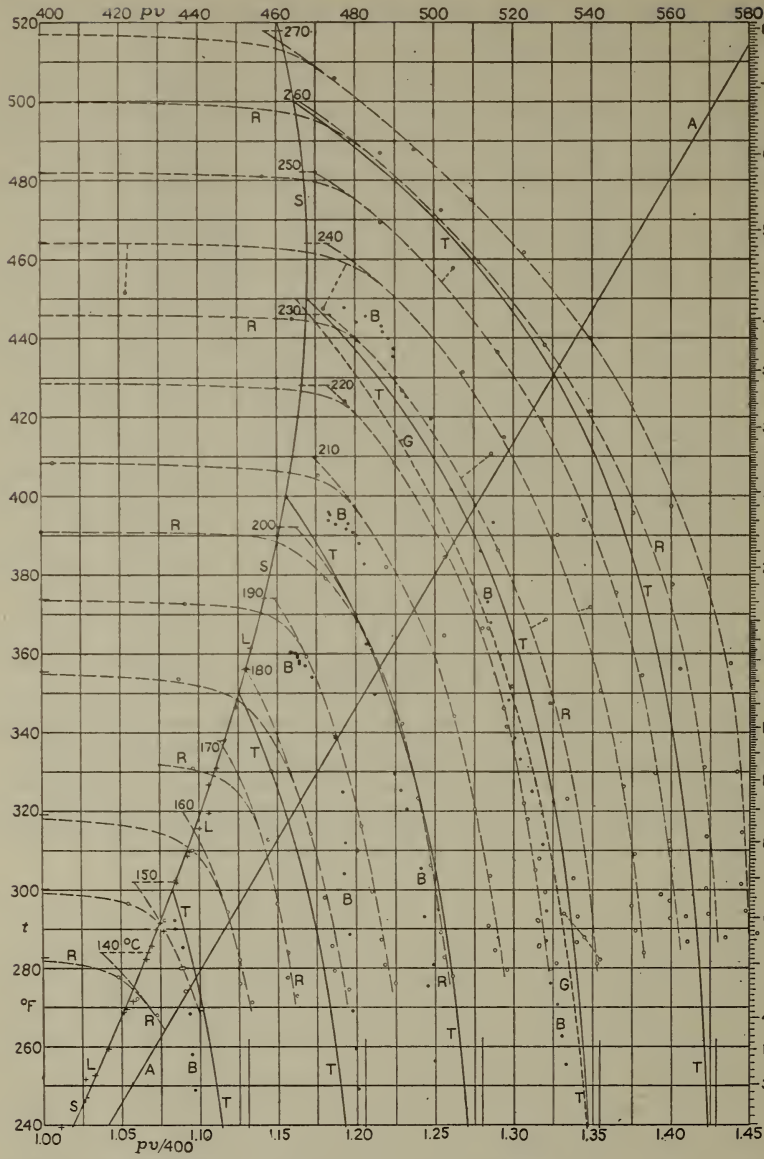

FIG. 16 PLOT OF pv ON p, WITH p TO VARIABLE SCALE CORRESPONDING WITH SATURATION TEMPERATURE, MADE TO TEST FORM OF $pv-p$ ISOTHERMALS.

CURVE *SS*, SATURATION LINE AS MODIFIED BY WRITER FROM LINDE'S EQUATION; THE MUNICH SATURATION VALUES, MARKED *L*, ARE PLOTTED ALONG THIS CURVE FROM 240 TO 360 DEG. FAHR.

FULL-LINE CURVES *TT*, ISOTHERMALS WITH LINDE VALUE OF F_p, AS WRITTEN IN EQUATION [D].

CIRCLE POINTS MARKED *R*, ISOTHERMAL OBSERVATIONS OF RAMSAY AND YOUNG, WITH DOTTED-LINE CURVES DRAWN AS PURELY EXPERIMENTAL RESULTS.

FIG. 16.—*Continued*

BLACK-DOTTED POINTS MARKED *B*, ISOTHERMAL OBSERVATIONS BY BATTELLI, EVIDENTLY OF INFERIOR ACCURACY.

ONE DOTTED CURVE AT 450 DEG., MARKED *G*, GOODENOUGH'S ISOTHERMAL, THE SAME AS CURVE *CE*—2 IN FIG. 10.

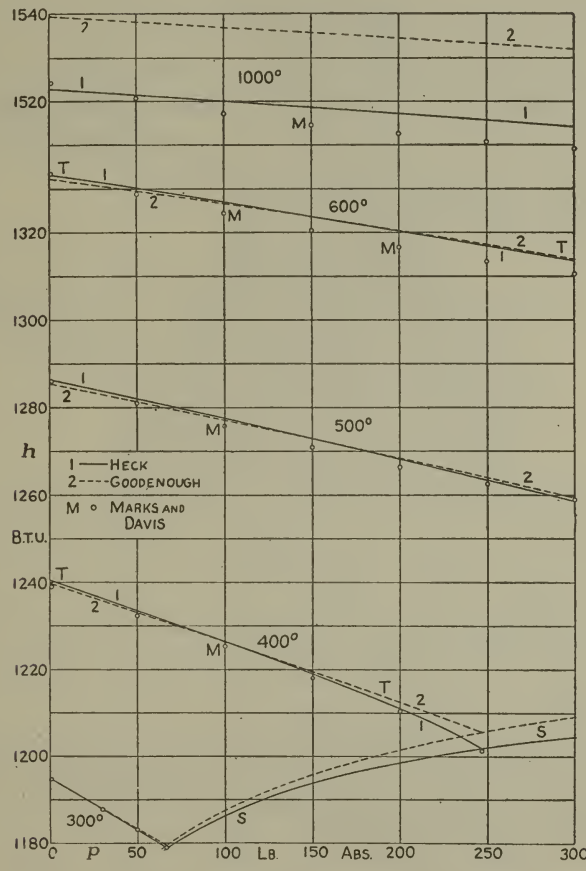

FIG. 17 PLOT OF TOTAL HEAT *h* ON PRESSURE *p*, FOR SUPERHEATED STEAM. CURVES *SS*, SATURATION LINES; CURVES *TT*, ISOTHERMALS AT THE TEMPERATURES MARKED. DIAGRAM IS CONTINUOUS UP TO 600 DEG., THEN SCALE IS BROKEN AND ISOTHERMALS AT 1000 DEG. ARE COMPARED.

a plot of total heat on pressure. A comparison of results is given in Fig. 17, where there is a wonderfully close agreement with Goodenough's formula for *i*, up to 600 deg. fahr. In the high range, represented by one set of isothermals at 1000 deg., a big discrepancy appears, due simply to the use by the writer of the Holborn-Henning

determination of heat added in superheating under atmospheric pressure. That this must be superseded by the higher result of Knoblauch and Mollier is probably true.

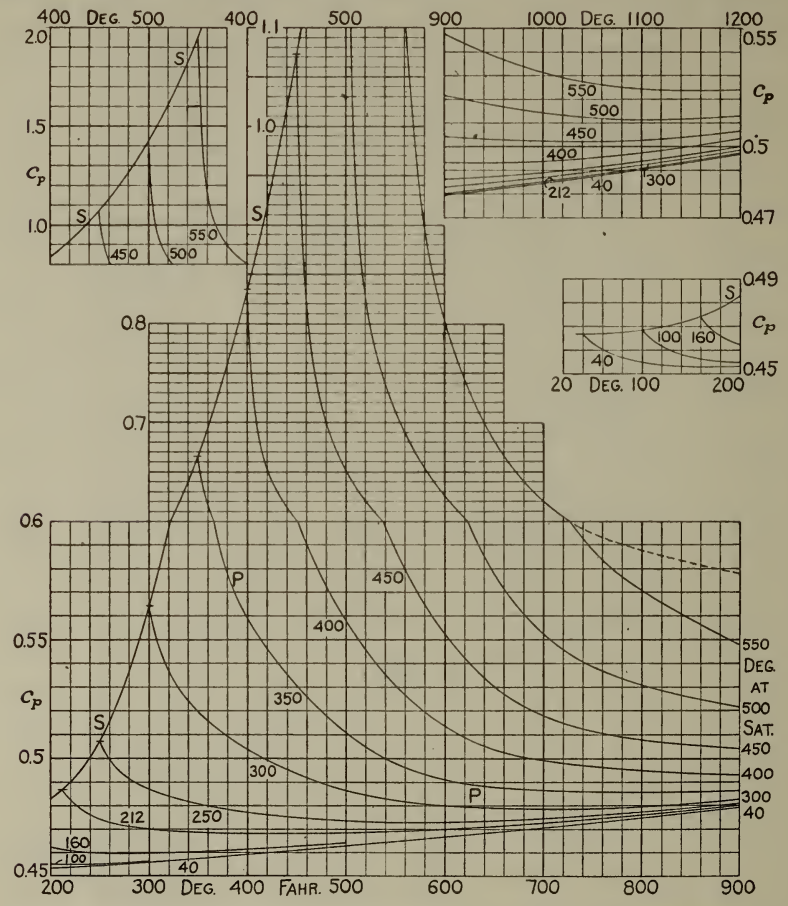

FIG. 18 CURVES OF SPECIFIC HEAT c_p. EACH PP CURVE IS DRAWN FOR THE PRESSURE CORRESPONDING WITH THE SATURATION TEMPERATURE WHICH DESIGNATES THE CURVE. THE SMALL SEPARATE SECTIONS COMPLETE THE MAIN DIAGRAM AT TOP, RIGHT, AND LEFT. THE SCALE OF c_p IS VARIED AS CONVENIENT. CURVE SS IS THE SATURATION LINE, AT WHICH SUPERHEATING BEGINS, AND ITS ORDINATE IS THE INITIAL HIGH VALUE $(c_p)_{sat}$.

Now the conformity of the isothermals in Fig. 17 exists in spite of the quite marked difference at saturation which appears in that figure and more extensively in Fig. 12. The writer's isothermals reach their lower saturation line by curving more rapidly as they

approach it. This involves the higher trend of the c_p curves toward saturation which is shown in Fig. 18. The line of $(c_p)_{sat}$, here very different from curve A in Fig. 7, is equally based on Planck's equation

$$(c_p)_{sat} = \frac{dH}{dt} - \frac{r}{T} + \frac{r}{u}\left(\frac{dv}{dt}\right)_p \quad \ldots\ldots\ldots\ldots\ldots [F]$$

The comparative numerical example presented in tabular form, Table 7, will effectively illustrate several important points.

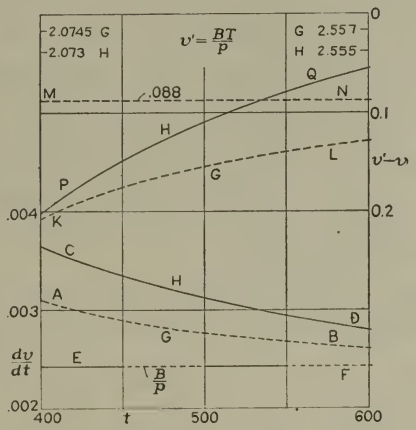

FIG. 19 RATE OF EXPANSION UNDER CONSTANT PRESSURE, AT THE PRESSURE 274 LB. ABS., WHICH CORRESPONDS WITH 400 DEG. SATURATION TEMPERATURE. CURVES *G*, GOODENOUGH'S FORMULA [*a*], CURVES *H*, LINDE'S LAYOUT AS SLIGHTLY MODIFIED BY THE WRITER.

In this example, the first line of each pair (case *a*) gives values computed from the formulae in the paper, with Marks' pressure-temperature equation, the second line (case *b*) gives results previously calculated by the writer. Detailed description of the items will be found in the subjoined notes. In lines 2 and 3, differences are insignificant; in lines 5, 6, and 10 appear discrepancies due to the disagreements shown in Figs. 10 and 12, becoming fairly large at the upper limit of temperature and pressure, or in the last two columns; but the really significant differences in data for this calculation are found in lines 4 and 12, or in the rates of change of volume and of total heat.

The general scheme of the calculation is to evaluate the last two terms of the expression in equation [*F*], then combine them in line 11, thus getting a number to be added to $\frac{dH}{dt}$. Note that up to 212 deg.

this "difference" is small and has little weight in comparison with the heat derivative; but in the high range relations are reversed, and the expansion rate $\frac{dv}{dt}$ assumes major influence.

Considering the low range first, in line 13*b*, $(c_p)_{sat}$ is seen to be a little greater at 32 deg. than at 120 deg., which is against reasonable probability; this indicates that the lower end of the writer's curve of total heat H ought to be swung a little way toward the right, perhaps by as much as 0.5 B.t.u. at 32 deg. Goodenough's layout of H and c_p appears to be self-consistent, but as already noted is wholly out of accord with the experiments on latent heat. The fancied need of extending c_p into the range of low temperature as a rectilinear function of t is not a sufficient reason for such radical departure from the trend of both direct and indirect determination of H. In Fig. 18 the lowest value of c_p, between 212 deg. and 32 deg., is 0.453, as against 0.432 in a curve for 1 lb. absolute pressure (102 deg. at saturation) from formula [*b*], given in Fig. 72 of Goodenough's "Principles of Thermodynamics."

In the high range, the whole question comes back to the characteristic equation, already considered from one point of view under that heading. Here the matter of interest is the rate of expansion $\frac{dv}{dt}$ under constant pressure, and comparisons are made in Fig. 19. Curves *AB* and *CD* show $\frac{dv}{dt}$, greatest at saturation and tending to approach the ideal value $\frac{B}{p}$ (in line *EF*). Curves in the upper group give the difference between ideal volume v' and actual volume v, measured in the diagram from top line down to curve: values of v' are marked on the figure for the limiting temperatures shown. Line *MN* represents the constant 0.088 in formula [*a*], and curve *KL* tends to approach this as superheat increases: curve *PQ* on the other hand, tends to become an asymptote to the top or zero line. As to the volumetric behavior of steam at high pressures and at very high temperatures we have no experimental data, and accurate experiments would be exceedingly difficult if not impossible. But the Linde concept seems to offer a better working hypothesis than the other.

The writer's curves of c_p in Fig. 18 have been spaced vertically by using a relation quite other than that expressed by the derivative

TABLE 7 COMPARATIVE CALCULATIONS OF SPECIFIC HEAT AT SATURATION, BY PLANCK'S EQUATION

1 $\frac{t}{T}$	32 491.6	120 579.6	212 671.6	320 779.6	400 859.6	480 939.6	560 1019.6
2 p	(*a*) 0.0886 (*b*) 0.0886	1.687 1.689	14.672 14.697	89.64 89.60	247.1 246.9	564.9 564.6	1130.5 1133.9
3 $\frac{B}{p}$	(*a*) 6.733 (*b*) 6.723	0.3535 0.3527	0.04064 0.04052	0.006652 0.006647	0.002413 0.002412	0.001056 0.001055	0.000527 0.000522
4 $\left(\frac{dv}{dt}\right)_p$	(*a*) 6.750 (*b*) 6.732	0.3598 0.3573	0.04327 0.04297	0.007774 0.00828	0.003093 0.00364	0.001513 0.00209	0.000876 0.001504
5 r	(*a*) 1079.7 (*b*) 1072.8	1027.6 1024.5	970.4 969.7	895.3 894.3	831.8 827.9	758.8 749.5	671.6 647.6
6 u	(*a*) 3309 (*b*) 3304	204.1 203.7	26.84 26.76	4.904 4.896	1.851 1.853	0.796 0.807	0.355 0.371
7 $\frac{r}{u}$	(*a*) 0.3263 (*b*) 0.3252	5.036 5.030	36.16 36.24	182.55 182.67	449.4 447.0	952.8 928.7	1893 1745
8 $AT\frac{dp}{dt}$	0.3246	5.032	36.23	182.95	447.3	925.0	1709
9 $\frac{r}{u}\frac{dv}{dt}$	(*a*) 2.203 (*b*) 2.189	1.812 1.797	1.565 1.557	1.419 1.514	1.390 1.616	1.442 1.940	1.658 2.626
10 $\frac{r}{T}$	(*a*) 2.196 (*b*) 2.182	1.773 1.768	1.445 1.444	1.148 1.147	0.968 0.963	0.808 0.798	0.659 0.635
11 diff.	(*a*) 0.007 (*b*) 0.007	0.039 0.029	0.120 0.113	0.271 0.367	0.422 0.653	0.634 1.142	0.999 1.991
12 $\frac{dH}{dt}$	(*a*) 0.411 (*b*) 0.464	0.398 0.432	0.359 0.371	0.287 0.265	0.219 0.166	0.121 0.02	−0.053 −0.43
13 $(c_p)_{sat}$	(*a*) 0.418 (*b*) 0.471	0.437 0.461	0.479 0.484	0.558 0.632	0.641 0.819	0.755 1.16	0.946 1.56

Line 1. Saturation temperature, at about equal intervals over the whole range of Figs. 10 and 12.

Line 2. Case (*a*), Marks' equation; case (*b*), writer's table.

Line 3. Writer used 0.5956 for B instead of 0.5963. This $\frac{B}{p}$ would be the same as $\frac{dv}{dt}$ if the steam were a perfect gas, following $pv = BT$.

Line 4. Case (*a*), from formula [*a*], differentiated as in Appendix No. 2. In any case, it is $\frac{B}{T}$ plus a corrective term. From the writer's table the latter is found. Compare equation [*C*] as

$$\frac{F_p}{p} \times \frac{\triangle F_t}{\triangle t} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [G]$$

Line 5. Cases (*a*) and (*b*) correspond with curves 2 and 1 in Fig. 12, using q as by curve H, Fig. 12.

Line 6. Values correspond with curves 2 and 1 respectively, in Figs. 10 and 11.

Line 7. In case (*a*) these are found by division of the preceding numbers, hence do not conform to Clapeyron's equation; in case (*b*) they are identical with the quantity represented by the heading of line 8.

Line 8. This ratio is from Marks' $p-t$ equation, for comparison with line 7*a* especially. Difference from 7*b* at 480 and 560 deg. measures the error in the writer's older $p-t$ relation.

Line 9. The last term of Planck's equation: $\frac{r}{u}$ is from line 7.

Line 10. Calculated with r as in line 5.

Line 11. The difference $\left(\frac{r}{u}\frac{dv}{dt}-\frac{r}{T}\right)$, which when added to $\frac{dH}{dt}$ will give $(c_p)_{sat.}$

Line 12. Case (*a*) is calculated from formula [*c*], using $\frac{dp}{dt}$ from Marks' equation.

Line 13. Add line 11 to line 12. Case (*a*) is represented by curve (*A*), Fig. 7

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of the rate of expansion; in fact there was a combination of several requirements which, with the data now existing, do not agree exactly in result, so that some adjustment and compromise was necessary. To bring the curves into accord with results obtained by applying Goodenough's method to the slightly modified Linde layout of v as related to p and t would call for some further adjustment. However, the difference between curves CD and AB in Fig. 19, notably the greater and more rapidly changing slope of CD, is all in the direction of a confirmation of the general form of Fig. 18.

CONCLUSIONS

There is and can be no question as to the validity of the principle expressed by the introductory and fundamental mathematical relation of the paper. The debatable matter is the quantitative validity of the characteristic or p-v-t equation. Tested by conformity to volumetric experiment and as to derived results affecting thermal relations, Goodenough's system does not seem so good as Linde's. The failure of equation [10] or formula [*c*] to agree at saturation with either experimental or derived heat values is a decided failure to represent all the data. The fact that curve 1 of Fig. 11, in accord with the latent heat experiments, lies within curve 2 which perhaps more exactly represents volume measurements, is some confirmation of the contention that all the latent heat determinations may be a

little small: but there is no justification for any such change of relative values as that from curves 1 and 3 to curve 2 in Fig. 13. On the whole, there is just a bit of ground for the opinion that the author has set up a mathematical system and tried to make the physical facts conform to it.

Forrest E. Cardullo. The tendency on the part of certain writers to regard thermodynamics as a branch of applied mathematics rather than a branch of physics is to be regretted. Pro-

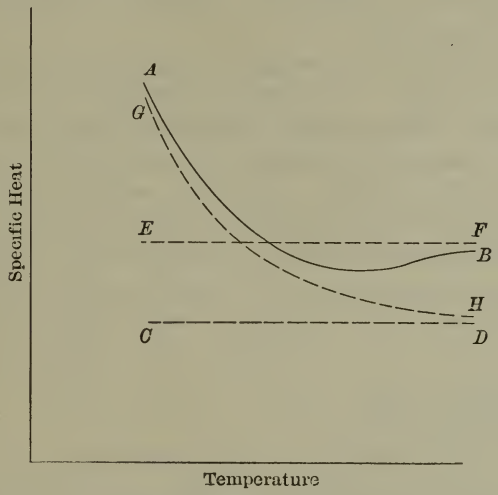

Fig. 20 Relation between Specific Heat and Absolute Temperature of Superheated Steam.

fessor Goodenough's paper is an extreme case in point. He first devises certain general equations of unknown form and then attempts to discover the form which these equations should have from a consideration of isolated investigations whose physical interpretation he does not attempt and for whose relative value he makes no allowance. It seems to me that this is approaching the problem from the wrong standpoint. A better method of procedure would be to frame a physical hypothesis which will account for the observed facts in the properties of superheated steam and then frame an equation which represents this hypothesis. The hypothesis must be founded upon our knowledge of the physical constitution of matter and the fundamental laws which govern it.

To illustrate, let us seek a physical interpretation of the curves which represent the specific heat of superheated steam at different pressures. One of these curves for steam of reasonably high pressure is represented by line *AB* in Fig. 20, in which the ordinates are specific heats, and the abscissæ absolute temperatures. From this figure it will be seen that the specific heat diminishes as the temperature increases, that it reaches a minimum value, and that it then slowly increases and apparently becomes asymptotic to the line *EF*. An appeal to the kinetic theory of heat in its application to imperfect gases will serve to explain the peculiar form of this curve.

A portion of the heat added is expended in increasing the translational energy of the steam molecules, i. e., in increasing the temperature. The quantity of heat so added is strictly proportional to the change in temperature and the effect of this item on the specific heat is constant. A second portion of the heat energy added is expended in increasing the internal energy of the molecules. Clausius has shown that the heat so added is proportional to the change in temperature and that the effect of this item on the specific heat is also constant. Were steam a perfect gas, these two items would be the only ones affecting heat, which would be constant, and represented in the diagram by the horizontal dotted line *CD*.

Since the steam molecules attract one another, it is necessary to expend a certain amount of energy in separating them as the steam expands at increasing temperatures. The further apart they are the less their attraction for one another and the less the energy required to separate them by a given distance. The amount of heat added for this purpose rapidly decreases as the temperature increases and the specific heat curve due to the combined effects already mentioned would follow the dotted line *GH*, which is asymptotic to the line *CD*.

The internal energy of the molecules varies. In some cases it has a high and in others a low value. When the internal energy of a steam molecule is so great as to overcome the mutual attractions of the atoms, it is split up into three parts, each one of which becomes a new monatomic molecule. The amount of heat necessary to raise the temperature a given amount is somewhat greater in the case of three monatomic molecules than it is in the case of one triatomic molecule. Consequently as the temperature of the steam increases the specific heat increases, since a larger

and larger proportion of the molecules is disrupted. Finally when all the molecules are disrupted the specific heat becomes asymptotic to the upper dotted line *EF*. The true curve of specific heat therefore lies slightly above the line *GH* and gets further and further above it as the temperature increases, having the form shown by the solid line and becoming asymptotic to *EF*.

Now it is an easy matter to compute the positions of *CD* and *EF*, to trace the curve *AB*, and finally obtain an empirical equation which will satisfy the conditions outlined.

It seems preferable, however, to compute the energy required to separate the molecules by methods analogous to those employed in celestial mechanics, and to compute the proportion of molecules disrupted at different temperatures by the theory of probabilities, and from the determinations so made to derive a rational equation for the heat of superheat of steam of any given pressure and temperature. It is then possible to derive the other properties of steam by recognized thermodynamic methods.

In employing this method the Joule-Thompson effect will be of great value, and since this is a quantity whose accurate measurement is comparatively easy, it seems that this method of attack offers the best solution of the problem.

Sanford A. Moss (written). I have made a comparison between lines of total heat as obtained from experiment and as computed according to Professor Goodenough's equations, and find fair but not exact agreement. This is shown in Fig. 21.

Peake[1], Grindley[2] and Greissman[3] have published results of experiments in which steam, supposedly dry and saturated, at a given high pressure was allowed to pass through an orifice to each of a number of lower pressures without doing any external work. The measurements of the low pressures and temperatures give lines of throttled expansion or lines of constant total heat.[4] These experiments have often been discussed[5] and it is agreed that they are fairly reputable. It seems certain that in each case the steam was originally somewhat wet so that the initial pressures and temperatures are certainly lines on the saturation curve and do not give the point of beginning of the lines of constant total

[1] Proc. Roy. Soc., 1905, vol. A 76, p. 185.
[2] Phil. Trans., 1900-1901, vol. 194 A, p. 1.
[3] Forschungsarb., 1904, vol. 13, p. 1.
[4] Trans. Am.Soc.M.E., vol. 30, pp. 736-739, discusses the theory.
[5] Proc. Amer. Acad. Arts and Sciences, 1910, vol. 45, p. 243.

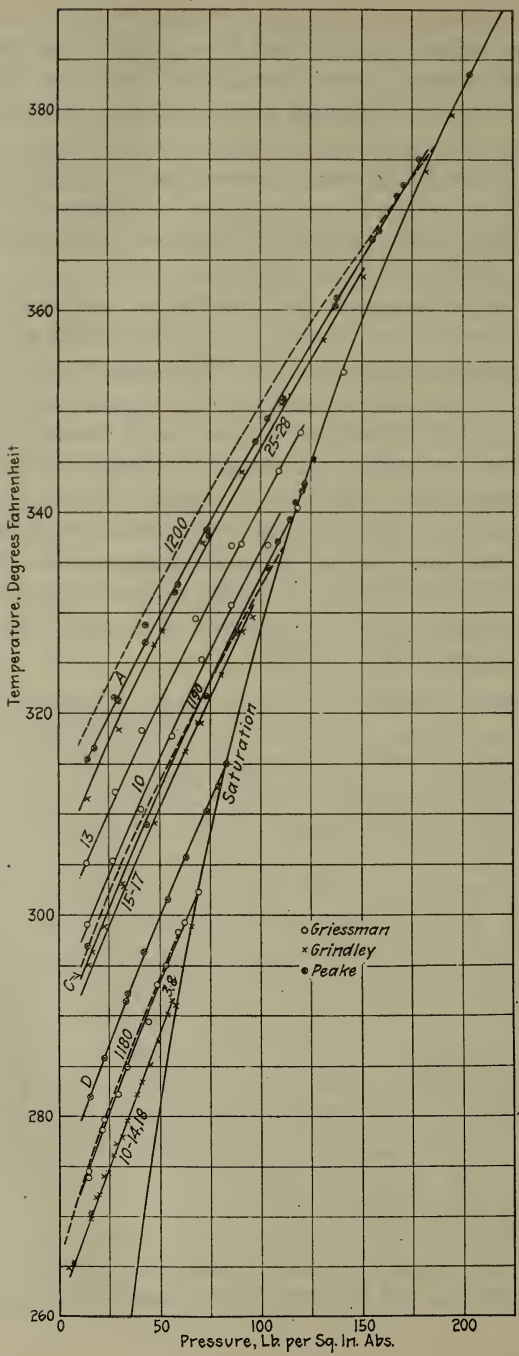

FIG. 21 THROTTLING CURVES ON LINES OF CONSTANT TOTAL HEAT

Full Lines Show Experimental Curves. Figures are Test Numbers. Dotted Lines Show Curves Computed from Goodenough's Equation. Figures are "Total Heat" B.t.u. per lb.

heat. In fact, in some cases (see Fig. 21) several of the early sets of readings of the condition of the steam beyond the orifice also lie on the saturation curve, indicating that the steam had not been quite dried out. For this reason, the lines of constant total heat in Fig. 21 are not continued to the saturation curve, owing to uncertainty as to where they would intersect. This intersection would come at some point below the lowest point of the same experiment on the saturation curve, but just how much below cannot be known. However, the fact that in the original condition the steam was not exactly dry does not vitiate the results in any way, as it is reasonable to assume that in each experiment the amount of moisture was exactly the same owing to the care taken in production of steam. The experiments, therefore, consist in throttling wet steam to each of a number of lower pressures. Since the initial quality is exactly the same in each case, all of the low points lie on the same line of constant total heat. Probably some of the irregularities of the points as plotted are due to slight differences in the amount of initial moisture and not to irregularities of observation of the low pressure and temperature.

These experiments were originally intended to give values of specific heat of superheated steam by computation from Regnault's values of total heat. This failed because of the now well-known errors of Regnault's total heat. These are the experiments which Dr. Davis[1] has used in obtaining his values of total heat of saturated steam by reversing the original purpose.

The values of "total heat" or "heat content," i, as given in Professor Goodenough's Table 2, were plotted against temperature for each of the given pressures and were found to give exceedingly flat curves so that temperatures for values of total heat intermediate between those given by Professor Goodenough can be obtained by linear interpolation. By this interpolation in Tables 1 and 2 successive values of pressure and temperature were found along three lines of constant total heat, the values of i being respectively 1180, 1190 and 1200. These values are given in Table 8. From these values the dotted lines in Fig. 21 were plotted. A large diagram was plotted containing on a single sheet all of the values of Peake, Greissman and Grindley as given in the

[1] Proc. Amer. Arts & Sciences, 1910, vol. 45, pp. 268-311; Trans. Am.Soc. M.E., 1908, vol. 30, p. 741.

tables in their papers. A line from each of the three experimenters nearest each of the dotted lines was selected and plotted in the accompanying figure. The points are all these values given in the original tables for each of the throttling curves. The full lines are smooth curves drawn through each set of points, taking cognizance, however, of neighboring points of the other observers. These lines differ slightly from the smooth curves given by the observers independently. Greissman represented his observations by lines with more curvature than are here given, while Peake used lines which were almost straight. The lines on Fig. 21 will be seen to represent the given points very

TABLE 8 COMPUTATION OF LINES OF CONSTANT TOTAL HEAT FROM GOODENOUGH'S EQUATIONS BY INTERPOLATION IN TABLES 1 AND 2

Total Heat = i	Pressure, Lb. per Sq. In.	Temperature, Deg. Fahr.
1200	Saturation	376.5
	150	365.8
	100	350.6
	50	332.6
	15	318.7
1190	Saturation	337.2
	100	332.2
	50	313.0
	15	296.9
1180	Saturation	301.8
	50	293.5
	15	275.4
	2	267.5

well. The coincidence of the slope and general shape of the lines obtained from the three entirely independent sources indicates that the slope and shapes as given are fairly accurate. It is in fact very remarkable that the three sets of observations should agree so well. The experiments have never before been brought together in the exact way shown in Fig. 21. However, Dr. Davis in his use of the three experiments independently in his total heat work has noted that they give the same final results.

It must be concluded, therefore, that the full lines in the accompanying figure very closely represent the true shape of the total heat curves for superheated steam and that Professor Goodenough's equation gives total heat fairly but not exactly.

THE AUTHOR. The constructive criticisms of Dr. Davis, Professor Heck and Mr. Moss are exceedingly valuable and suggestive. The various defects in the theory pointed out in the course of the discussion are freely conceded, and the author is not inclined to defend results that are palpably not in accord with well-established conclusions.

In the light of the various criticisms offered, the whole subject has been reviewed since the publication of the paper, and it seems possible at the present writing that the theory may be so modified as to remove in large measure the objections urged against it. Mr. S. L. Simmering, a graduate student in the University of Illinois, has undertaken a revision of the characteristic equation and he has succeeded in establishing a new equation that meets all requirements more nearly than either the Linde equation or the equation proposed in the paper. The principal change is the reduction of the exponent n. With the new equation the criticism of Dr. Davis in connection with Fig. 9 is obviated; the reduced values of c_p for various pressures show no tendency to separate. An arbitrary change has been made in the function $\psi(T)$. Originally it was given the linear form $\alpha+\beta T$; the new form is

$$\alpha+\beta T+\frac{\gamma}{T^2}.$$

While the work is not completed, sufficient data are available to justify the following statements:

a The new equation will give saturation values of the volume v almost identical with the Marks and Davis values throughout the range 32 to 400 deg. fahr.

b The values of c_p will agree substantially with the results of the Knoblauch and Mollier experiments.

c The saturation values of the total heat will agree with the Marks and Davis values throughout the range 32 to 400 deg. fahr. with a maximum difference of about 1 B.t.u.

d The calculated values of latent heat and volume will substantially satisfy the Clapeyron relation, thus removing the serious objection urged by Professor Heck.

e The formula for total heat will be more suitable for extrapolation to temperatures above 400 deg. Thus at the critical temperature, it gives about 1000 B.t.u. instead of 1170 B.t.u.

f It is probable that the upper limit of superheat, namely,

400 deg. cent., set by Dr. Davis, may safely be raised to 700 or 800 deg. cent.

The final results of the revision will be communicated to the Society in due time.

Professor Cardullo's discussion hardly demands a reply. Of the method of investigation proposed by Professor Cardullo, Planck says (preface to Treatise on Thermodynamics): "Obstacles, at present unsurmountable, however, seem to stand in the way of its further progress. These are due not only to the highly complicated mathematical treatment, but principally to essential difficulties in the mechanical interpretation of the fundamental principles of Thermodynamics."

No. 1357

THE REDUCTION IN TEMPERATURE OF CONDENSING WATER RESERVOIRS DUE TO COOLING EFFECT OF AIR AND EVAPORATION

By W. B. Ruggles, New York
Member of the Society

The tests described in this paper were made with the idea of determining if possible, a fairly reliable factor by the use of which the necessary size of a cooling reservoir for condenser water could be predetermined for any assumed power and weather conditions.

2 In 1908, while designing the mill for the Crescent Portland Cement Company at Wampum, Pa., it was found necessary to install a cooling device for the condensing water from the engines. The mill stands on a level plateau about 110 ft. above the Beaver River and to pump water from the river for condensers meant a considerable expense for power. On the side of the plateau furthest from the river was a natural depression alongside the Erie division of the Pennsylvania Railroad which offered a desirable location for a reservoir (Fig. 1). After a diligent search, no data were found as to how large a reservoir had to be to give the necessary cooling effect to the water, so that it might be used continuously. A dam, however, was built 275 ft. long and 18 ft. high, of skeleton reinforced concrete construction which would impound about $6\frac{1}{2}$ acres of water (Fig. 2).

3 Three tests were made during the year 1911, each of one week's duration in order to determine the heat radiation from the surface of the reservoir. These tests were made in May when the temperature was moderate, in July when the temperature was high and in November when the temperature was low. Readings were taken of the temperature of the river water, the intake water to the power house, the tail water from the condenser and of the air. The vacuum,

Presented at the Spring Meeting, Cleveland 1912, of The American Society of Mechanical Engineers,

the power, the amount of water pumped from the river to the reservoir, and the rainfall were also recorded. These data were taken by James Toler, chief engineer of the Crescent plant and by the writer's assistant, John E. Mason. Unfortunately the temperature of the rain was not taken, but it has been assumed the same as the temperature of the air, so that the figures for that are only approxi-

FIG. 1 VIEW OF THE CRESCENT PORTLAND CEMENT COMPANY'S PLANT SHOWING RESERVOIR

mate, but a difference of 10 deg. in the temperature of the rain would not alter the result by more than $\frac{2}{10}$ per cent.

4 The engines used in the power house are three Ball and Wood compound condensing engines, 26 in. by 52 in. by 36 in. The air compressors are two compound Laidlaw-Dunn-Gordon compressors, the steam cylinders of which are 12 in. and 21 in. in diameter respectively, the air cylinders $11\frac{1}{2}$ in. and 24 in. respectively, the stroke 24 in. The steam from all of these is condensed in an Alberger barometric condenser equipped with a dry vacuum pump and the circulating water supplied by a direct-connected centrifugal pump driven by a synchronous motor; an engine-driven circulating pump and a second dry vacuum pump being installed as reserve.

5 The water from the condenser flows about 50 ft. through a tile pipe into the east side of the reservoir about 100 ft. from the north end. A dike was built south of this inlet to the reservoir extending about 50 ft. toward the center of the reservoir and then

north nearly to the north end. This compels the circulating water to flow up and around the end of this dike and down toward the dam, a distance of about 1100 ft., before it reaches the intake to the power house.

6 Water to supply the reservoir is pumped from the river by a centrifugal pump direct connected to a 150-h.p. motor. The pump delivers an average of 2324.25 gal. of water per minute and it is found that about 8 hours pumping per week is sufficient to make up for the evaporation and seepage in the reservoir. The water from the river pump is delivered into an open well situated on top of the plateau where the mill is located. It flows by gravity from

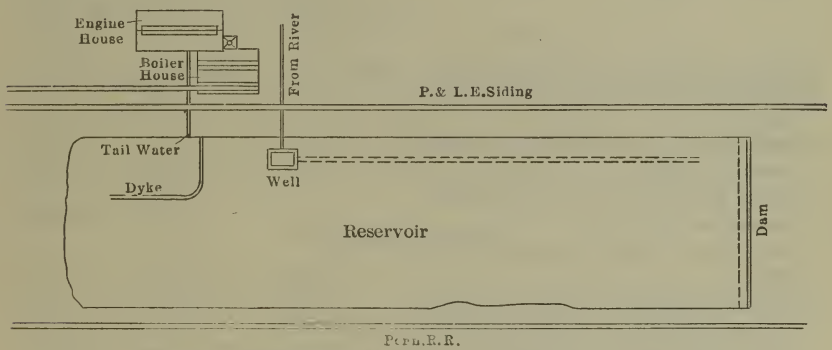

FIG. 2 DIAGRAM SHOWING 6½-ACRE RESERVOIR AND CONNECTIONS

this well through a 24-in. tile pipe into another concrete well, located in the reservoir about 800 ft. north of the dam. From this well it flows through a No. 24 tile pipe laid on the bottom of the reservoir to within 6 ft. of the dam. The intake water for the boiler feed and condenser is drawn from this well so that the water must flow up through the concrete tile pipe from the bottom of the reservoir at its deepest point. While water is being pumped from the river, there is a flow south through this tile pipe and when not pumping, the water is drawn in the opposite direction through it.

7 The average horsepower developed in the engines during the 21 days of these tests was 2446. Incidentally, it may be stated that there was an average of 3062 bbl. of cement manufactured per day during the same test, or 1 bbl. of cement for 0.8 h.p.

8 The results of the tests during these three weeks are as uniform as could be expected; the heat lost per square foot of surface being slightly less per 1 deg. difference of temperature in cold weather than

in warm weather. This difference is undoubtedly due to an increased amount of evaporation during the warmer period.

9 The average humidity for the different weeks during which the tests were made was 58.5 per cent in May, 62.3 per cent in July and 71.2 per cent in November.

10 Table 1 gives the results in detail of the tests made.

TABLE 1 TEST OF HEAT RADIATION FROM SURFACE OF CONDENSER WATER RESERVOIR AT PLANT OF CRESCENT PORTLAND CEMENT CO., WAMPUM, PA.

Area of reservoir, 288,000 sq. ft.; average depth of reservoir, 5.36 ft.; capacity of reservoir, 1,543,680 cu. ft. = 96,480,000 lb.

Date of Tests	Week ending May 7, 1911	Week ending July 12, 1911	Week ending Nov. 27, 1911.
Amount of water pumped from river, lb.	10,458,956	29,050,875	4,648,140
Average temperature of river water, deg. fahr	57.5	77	36
Average temperature of intake to power house, deg. fahr	72.75	91.43	61.71
Average temperature of tail water from condenser, deg. fahr	101.36	129.43	90.71
Average temperature of reservoir, deg. fahr	87.05	110.00	76.71
Average temperature of air, deg. fahr.	51.00	78.43	33.30
Average difference of temperature between water and air, deg. fahr	36.05	31.57	43.41
Change in temperature of reservoir during test, deg. fahr	0.25	7.00	2.00
Three 26 in. by 52 in. by 36 in. compound condensing engines, h.p-hr	397,807	411,320	426,936
Two 12 in. by 21 in., 11½ in. by 24 in. by 24 in. air compressors, h.p-hr	46,536	46,536	46,536
Average vacuum at 30 in b arometer, in.	26.4	22.2	26.6
Average water consumed by engines, per h.p. per hr., lb	14.46	15.64	14.41
Average water consumed by compressors per h.p. per hr., lb	18.85	20.12	18.83
Steam condensed by engines, lb.	5,752,289	6,433,045	6,145,148
Steam condensed by compressors, lb.	877,204	936,314	876,273
Latent heat of steam condensed, B.t.u. per. lb	1024.7	1007.1	1026.0
Heat delivered to reservoir by engines, B.t.u	5,894,370,000	6,478,720,000	6,304,922,000
Heat delivered to reservoir by compressors, B.t.u	898,871,000	942,961,000	899,056,000
Heat to raise river water to average temperature of reservoir, B.t.u.	309,062,000	958,679,000	189,226,000

TABLE 1—*Continued*

Date of Tests	Week ending May 7, 1911	Week ending July 12, 1911	Week ending Nov. 27, 1911
Heat given up or retained in reservoir during test, B.t.u.	24,120,000	675,360,000	192,960,000
Heat reduction in reservoir due to rain, B.t.u.	21,630,000	56,700,000	46,200,000
Heat absorbed by air and evaporation during seven days, B.t.u.	6,438,429,000	5,730,942,000	6,564,509,000
Heat absorbed by air and evaporation. per sq. ft. of surface seven days, B.t.u.	22,356	19,899	23,495
Heat absorbed by air and evaporation per sq. ft. per hr., B.t.u.	133.1	118.4	139.8
Heat absorbed by air and evaporation per sq. ft. per hr., per 1 deg. difference, B.t.u.	3.69	3.71	3.22

11 It appears from the tests that under ordinary conditions, in the northern part of the United States with engines using 15 lb. of water per h.p-hr. and a vacuum of 26 in., a reservoir having surface of 120 sq. ft. per h.p. would be ample for cooling the condensing water.

DISCUSSION

Forrest E. Cardullo. It may be shown from the kinetic theory of heat that the rate of evaporation from a water surface is approximately proportional to the difference between the saturation pressure of water vapor at the temperature of the water and the actual pressure of the water vapor in the air. Experiment shows that the rate of evaporation in still air, i. e., air in which the only currents are those caused by the temperature conditions and not by the wind, the rate of evaporation is approximately 0.3 lb. of water per sq. ft. of surface per hour for every pound difference in vapor pressure. When there is an appreciable amount of wind, the rate of evaporation becomes higher, and in case the wind is very brisk it may become four or five times as great as the value given. Furthermore, the rate of evaporation depends on the size and form of the pond, being greater in the case of a small and narrow pond and less in the case of a large or circular pond, since the air which sweeps across the large pond will be partially saturated with moisture before it has gone very far.[1]

[1] For the development of this theory and its application to the design of cooling ponds see Practical Thermodynamics, McGraw-Hill Book Co., pp. 264-267.

There is a reasonable concordance between theory and the actual results which Mr. Ruggles obtains. For the week ending May 7 the amount of heat carried away was about 15 per cent greater than theory would indicate. For the week ending July 12 it was a few per cent less, but for the week ending November 27 it was nearly twice as great.

This is not contrary to theory, as it may seem to be, since one very essential element has been omitted in the data of Mr. Ruggles' paper, namely the velocity and direction of the wind, and it is reasonable to assume that in November the amount of wind would be considerably greater than during May or July. The direction of the wind would also have a considerable effect, since a wind blowing the long way of the reservoir would produce a lower rate of evaporation than one which blew crosswise.

Mr. Ruggles gives for the average temperature of the reservoir the mean of the temperature of the intake and the condenser discharge. As a matter of fact, it could be but little higher than the temperature of the intake. He assumes that the steam rejected by the engines was of 100 per cent quality. The quality was probably not far from 85 per cent and might be as low as 80 per cent. His figures on the amount of heat delivered to the reservoir are therefore about 20 per cent too high.

An inspection of Mr. Ruggles' paper would lead one to expect that a cooling pond would dispose of 3½ B.t.u. per sq. ft. per hr. per deg. difference in temperature between the water in the pond and the air. When, however, the temperature of the condenser intake is taken for the temperature of the pond, and a quality of 84 per cent is assumed for the steam discharged to the condensers, the B.t.u. per sq. ft. per hr. per deg. difference in temperature ranges from 4.13 to 7.60, being higher in hot weather than in cold weather. This is as it should be, since the relative humidity is less and the vapor pressure difference and rate of evaporation are greater in hot weather.

The average rate of evaporation for the week is not the rate of evaporation which would be produced by the average conditions of temperature and humidity for the week. On this account, and because the weather cannot be predicted in advance, and on account of the great variations of weather conditions to be expected, it is foolish to attempt any degree of refinement in the design of cooling ponds. It is not desired to design a cooling pond which will give an intake temperature of 90 deg. at

2 p.m. on July 4, but to design one which will, when used in connection with the other apparatus of the station, give the required power output at a minimum total cost. Either the data given by Mr. Ruggles or the method mentioned in this discussion will enable one to do this.

Wm. T. Donnelly said he had discussed the elements of the subject with the author before the design of the reservoir and afterwards had discussed the results. It seemed to him that more depended upon the humidity of the air than upon any other single factor. The atmosphere must be considered as a condenser and the rate of flow of vapor into the atmosphere is affected very much by the condensation.

In the winter the greater evaporation is due to the relatively lower humidity of the air, and the passing off of the vapor from the warm surface is more rapid on that account. The author is to be commended for giving information along new lines dealing with manufacturing conditions as met with all over the country, and it is not necessary to have it to the third or fourth decimal place to make it useful.

Willis H. Carrier. As Professor Cardullo suggested, it would seem that the rate of heat transfer would be in direct proportion to the total heat difference in the air at the air temperature, and at a saturated air temperature, corresponding to the water temperature of the air in the pond. In other words, there is a certain amount of heat in the natural air to start with, which being in immediate contact with the pond is raised to and saturated at the temperature of the pond, so that the amount of heat taken away from the pond under the same conditions of wind would be in direct proportion to this total heat difference. This would naturally lead to the conclusion that there would be a much higher rate per degree difference at the high temperatures as the total heat of saturated air increases very rapidly at the higher temperatures. But this is not so, and Professor Cardullo has offered an explanation as to the reason. From an engineering standpoint it is very interesting to know that the rate of heat transfer is much greater in the months when the air is cooler and the temperature of the pond is lower, so that, as Professor Cardullo says, calculations may be made upon the summer conditions with very little wind.

There is also another point to be taken into account where there is no forced circulation, that is, that when the temperature increases, the rate of transmission would also be increased

TABLE 2 EFFECT OF WET AND DRY BULB TEMPERATURES UPON THE CAPACITY OF AIR TO COOL WATER

Air Temperature			A_2-A	Water Temperature		W_1-W_2	$\frac{A_2-A_1}{W_1-W_2}$
Entering Wet-Bulb A	Entering Dry-Bulb	Leaving A_1		Entering W_1	Leaving W_2		
TEST NO. 1, ½-IN. SPRAYS							
75	80	95.7	20.7	110	93.6	16.5	1.26
75	78	96.8	21.8	110	94.0	16.1	1.35
75	85	96.0	21.0	110	93.6	16.5	1.28
75	90	95.7	20.7	110	93.2	16.9	1.23
75	95	94.7	19.7	110	92.1	18.0	1.09
75	100	96.4	21.4	110	93.1	17.0	1.26
75	105	95.7	20.7	110	93.6	16.5	1.26
75	100	95.8	20.8	110	92.6	17.5	1.19
TEST NO. 2, ⅜-IN. SPRAYS. CONDITIONS AS IN TEST NO. 1							
75	77	91.5	16.5	110	88.9	21.2	0.78
75	80	92.0	17.0	110	89.4	20.7	0.82
75	80	91.5	16.5	110	88.4	21.7	0.76
75	90	91.5	16.5	110	88.5	21.6	0.76
75	95	91.3	16.3	110	88.4	21.7	0.75
75	100	91.3	16.3	110	88.4	21.7	0.75
75	105	91.5	16.5	110	88.4	21.7	0.76
TEST NO. 3, ½-IN. SPRAYS. CONDITIONS REVERSED. WATER USED TO COOL AIR							
80	81	61.1	−18.9	58.8	64.4	−5.6	3.38
80	84	60.8	−19.2	58.8	64.4	−5.6	3.43
80	87	61.1	−18.9	58.8	64.3	−5.5	3.44
80	84	60.9	−19.1	58.8	64.3	−5.5	3.47
80	90.5	60.9	−19.1	58.9	64.1	−5.2	3.68
80	93.5	60.8	−19.2	58.8	64.1	−5.3	3.62
80	96.0	60.7	−19.3	58.8	63.9	−5.1	3.79
80	98.0	60.8	−19.2	58.9	64.0	−5.1	3.77
80	104.5	60.7	−19.3	58.9	63.9	−5.0	3.86
80	107.0	61.2	−18.8	58.9	64.0	−5.1	3.69

by the fact that the natural air currents produced by this temperature are increased, producing an induced circulation. Just what the effect of this is and what the laws governing it are is

very difficult to determine, but probably they are very similar to those of direct radiation, which would give a greater rate of transfer at a greater temperature difference. This, however, does not seem to be of much interest from a practical standpoint. So many variables enter into the conditions that the approximate constants as shown by the results of these tests may be accepted as the average conditions.

Mr. Carrier referred to the statement by Mr. Donnelly with reference to the effect of humidity upon the rate of heat transferred. The question is rather one of wet bulb temperature which is the measure of the total heat in the air and is the temperature at which water will arrive if subjected to the air for a sufficient time.

He had recently made an extensive series of experiments which bear directly on the point raised. They show that the effect of air in cooling water in towers or ponds, other conditions remaining constant, depends entirely on the wet-bulb temperature of the air, and that the variation in dry-bulb temperature and in relative or absolute humidity has in itself no effect, except in so far as it modifies the wet-bulb temperature, or temperature of evaporation as it may more properly be called. The theoretical reason for this was shown in the writer's papers on Air Conditioning[1], but these tests prove this theory indisputably. While this fact is of prime importance in the design of cooling towers and ponds it has apparently been overlooked both by American and European engineers.

These tests were made with a spray type of cooling tower, using different sizes of spray nozzles. In the first two air was used to cool the water, while in the third the process was reversed, the water being used to cool and dehumidify the air. The test conditions and results attained are shown in Table 2. In all cases the wet-bulb temperature was maintained constant during the test, while the dry-bulb temperature and moisture contents was varied through quite a wide range. The water temperature entering as well as the water quantity and air velocity were maintained constant. As a result it will be seen in each case that the leaving air and water temperature remain constant to a remarkable degree.

[1] Trans. Am. Soc. M. E., vol. 33. Rational Psychrometric Formulae, p. 1005, and Air Conditioning, p. 1055.

THE AUTHOR. There is no doubt, as Professor Cardullo states, that the direction and force of the wind have considerable influence on the amount of evaporation, but as these tests were made in order to obtain some practical and reasonable basis for estimating the size of a reservoir required for any particular plant and not for a mere academic discussion of theories, the amount of wind was purposely omitted as introducing an uncertain factor which would be of no future use. Moreover the author considered that the average of three different widely separated weeks would give a fair average of yearly conditions.

Professor Cardullo is mistaken regarding the average temperature of the pond being nearly the temperature of the intake, and the author cannot understand where he could get such information. Actual readings 100 ft. apart show the mean temperature, as he stated, to be the average between the tail water temperature and that of the intake water.

Presumably Professor Cardullo in referring to the quality of the steam being only about 80 per cent or 85 per cent meant the percentage of dryness and probably had in mind the steam as it left the low-pressure cylinders or as it entered the condenser. But as the amount of steam figured was that which entered the engine and was corrected by calorimeter test for moisture, and as all the heat, except that utilized for power, went into the condenser and reservoir and a very small percentage was due to radiation from the engine cylinders. his criticism of the tests in this respect is futile.

In making these tests the author had no idea that the results obtained would be absolute, but thought that they would throw light on a subject which as far as he was able to learn had not been treated before.

No. 1358

RESULTS OF TESTS ON THE DISCHARGE CAPACITY OF SAFETY VALVES

By E. F. Miller, Boston, Mass.

Member of the Society

The tests quoted in this article (Tables 1 and 2) were made during the spring and summer of 1910 for the information of the Crosby Steam Gage & Valve Company, by whom permission has been given to present the contents of the report to the Society, believing that the results will be of some value to its members.

2 In all the tests the valves were without springs and were set a definite distance from their seats and held rigidly during the tests. They are shown in Figs. 1, 2 and 3 as modified for these tests. As a result of the tests the locomotive valve is now made as shown in Fig. 4. In order to avoid unequal expansion, the metal of the valve body and the metal of the spindle were made the same in the case of the locomotive valves; difference of expansion was similarly obviated as far as possible in the inspector valves. The spring was replaced by a spindle attached to the valve. This spindle extended through the top of the valve and carried at its free end a pointer which traveled over a graduated cylinder. The lower part of the spindle was threaded with a 20 thread. The graduated cylinder at the top was divided into five parts, each representing 0.01 in. A motion of the pointer of approximately 1 in. corresponded to a 0.01 in. lift of the valve from its seat. Tests on the 3-in. valves were run with lifts set at 0.10,0.08, 0.05 and 0.02. Tests on the $3\frac{1}{2}$-in. valves were run with the same lifts at 100 and 150 lb. pressure, but the capacity of the condenser prevented running the $3\frac{1}{2}$-in. valves with 0.10 in. lifts at 200 lb. pressure.

3 The valves were connected with the boilers through a line of 5-in. pipe, into which two Babcock & Wilcox boilers of 500 boiler h.p. discharged. The 5-in. pipe connected with a 10 ft. length of 10-in.

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pipe on the end of which was a blank flange 2 in. thick. To this flange the safety valve under test was bolted (see Fig. 5). This flange was bored on the bottom with a hole considerably larger than the inlet to the valve. The entrance edge of this hole was rounded with a curve of 1 in. radius. On the other side of this blank flange and enclosing the safety valve was a 10-in. flanged tee. The outlet

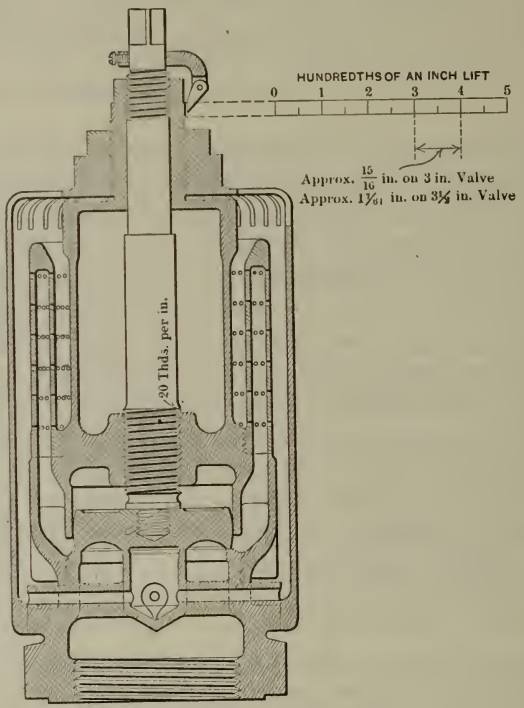


FIG. 1 CROSBY MUFFLED LOCOMOTIVE POP SAFETY VALVE

of this tee led to the condenser. The end of the tee on the straight run was covered with a blank flange. Between tests this blank flange on the end of the tee was removed and the setting of the valve changed. After each test the valve was examined to see if it had moved from its previous setting. The steam passing through the valve was condensed in a surface condenser vented to the air through a $2\frac{1}{2}$-in. pipe. The condenser was tested immediately before and after each test and was found to be absolutely tight. The condensed steam was pumped to two weighing tanks, each of about 1500 lb. capacity.

4 The steam was led from the boiler through a small separator

in the boiler room, then through a second separator located about 20 ft. from the safety valve. A calorimeter was attached to the 10-in. pipe near the safety valve and the quality of the steam determined. In all of the tests the steam was practically dry.

5 To see whether or not there was any pressure in the muffler or in the outer valve casing, a copper pipe was connected and led through a stuffing box outside of the 10-in. tee and a low-pressure

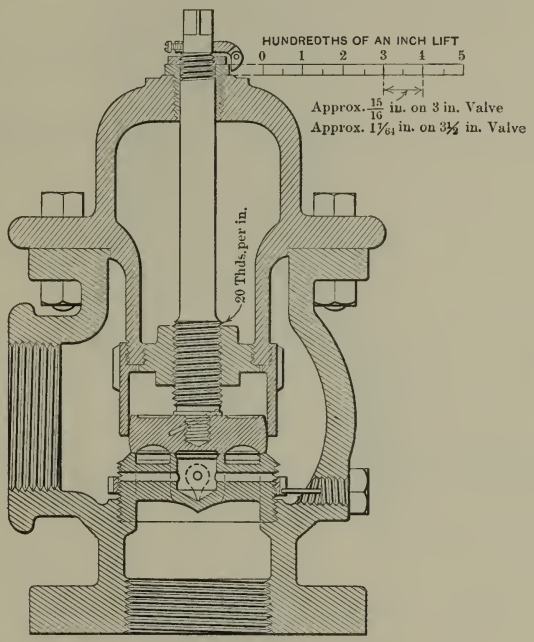


Fig. 2 Crosby Flat Seat Inspector Pop Safety Valve

gage attached. The boiler gage used was a Crosby standard test gage attached to the 10-in. pipe. In nearly all of the tests this gage was read at 1-minute intervals. In recording the pressures averages were taken from five readings in each case.

6 The pressure coming on the valve tends to make the opening through the valve greater than the "lift as set," due to the yielding of the metal. To determine this yielding, the valves were placed in a small Riehlé testing machine and a load equal to the steam pressure on the bottom of the valve applied to the valve. The additional opening due to this yielding was determined by micrometer measurements.

7 All tests with low lifts were made at least 60 minutes long to minimize any error due to a difference of level in the hot-well under the condenser.

8 The condensed steam is pumped out of the hot-well by a pump operated through a float and valve. A difference of about 1 in. is required ordinarily to make this float operate. One inch difference in level in the hot-well means an error of 9 lb. of water.

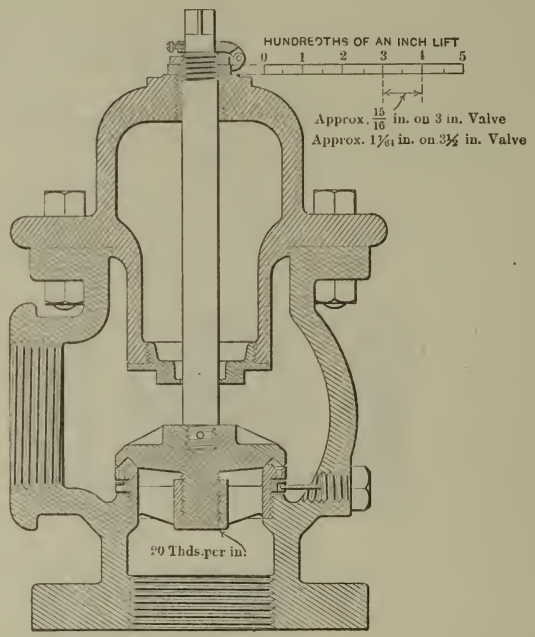


FIG. 3 CROSBY BEVEL SEAT INSPECTOR POP SAFETY VALVE

9 The different results are given both in tabular form (Tables 1 and 2) and in the shape of plots (Figs. 6 and 7). The different tests on one valve have been reduced to a common pressure by assuming the discharge through a given orifice to be proportional to the absolute boiler pressures. For such small variations in pressure as there were in these tests this assumption can introduce no error. Tables 1 and 2 give summaries only of results obtained in the tests. Tables were prepared, however, giving the results in detail, an example of which is shown in Table 3 which gives a part of the detailed records[1] summarized in Tables 1 and 2.

[1] Other records may be consulted in the rooms of the Society.

10 In the locomotive valves the muffler casing was screwed down to its lowest position. In the inspector valve flat seat the ring was

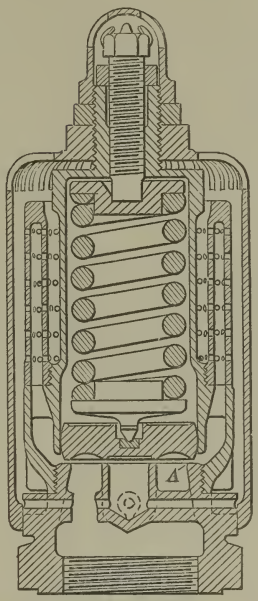


FIG. 4 SECTIONAL VIEW OF CROSBY MUFFLED LOCOMOTIVE POP SAFETY VALVE AS NOW MADE

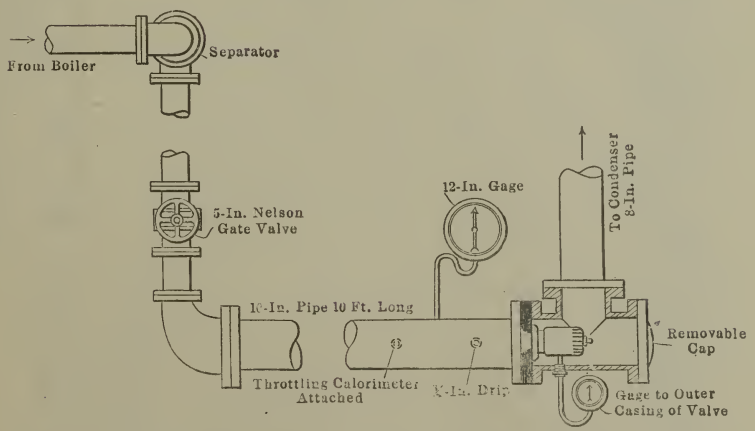


FIG. 5 ARRANGEMENT OF PIPING FOR SAFETY-VALVE TESTS

screwed down so as to leave the holes half open. There was no ring on the bevel-seated valves.

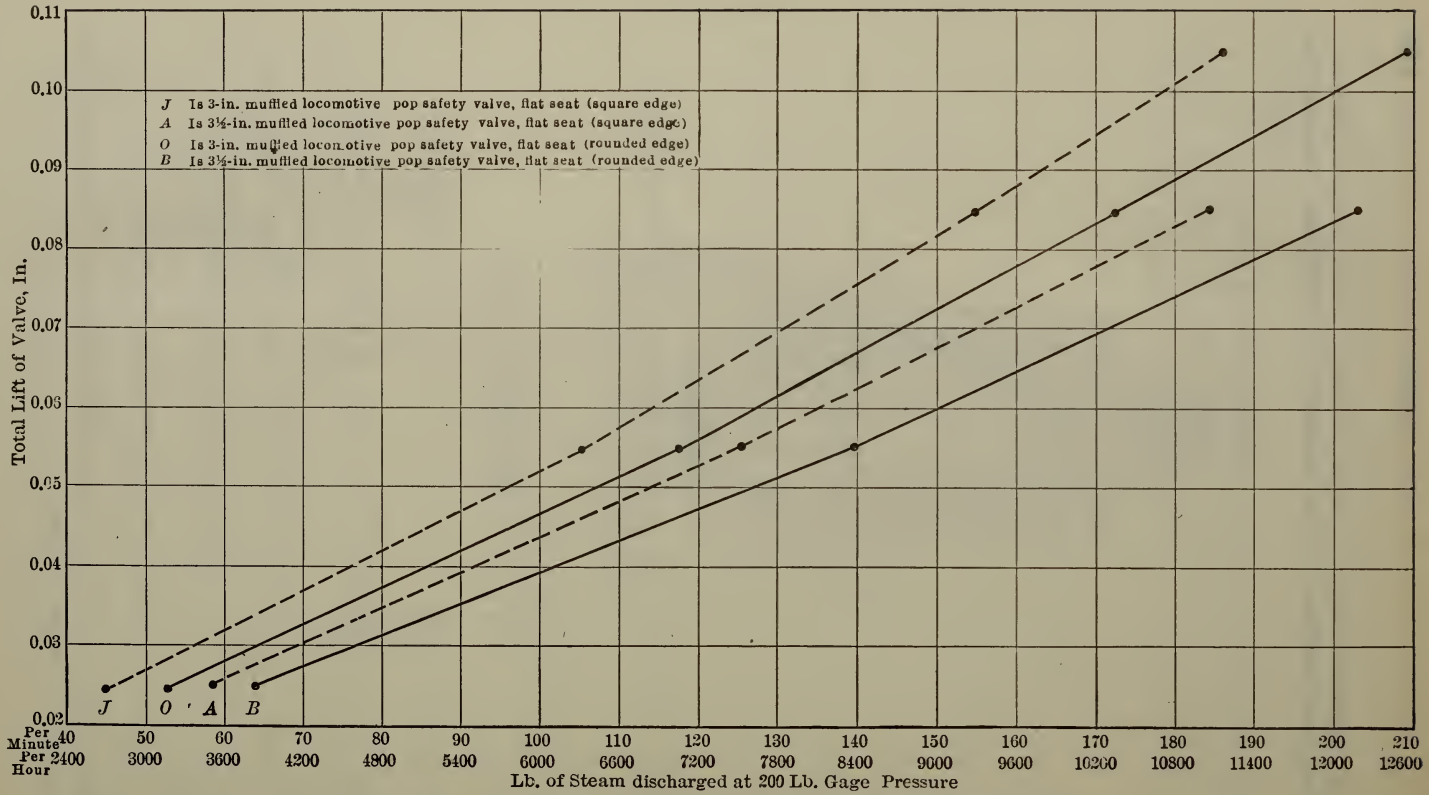


FIG. 6 RESULTS OF TESTS ON CROSBY LOCOMOTIVE POP SAFETY VALVES

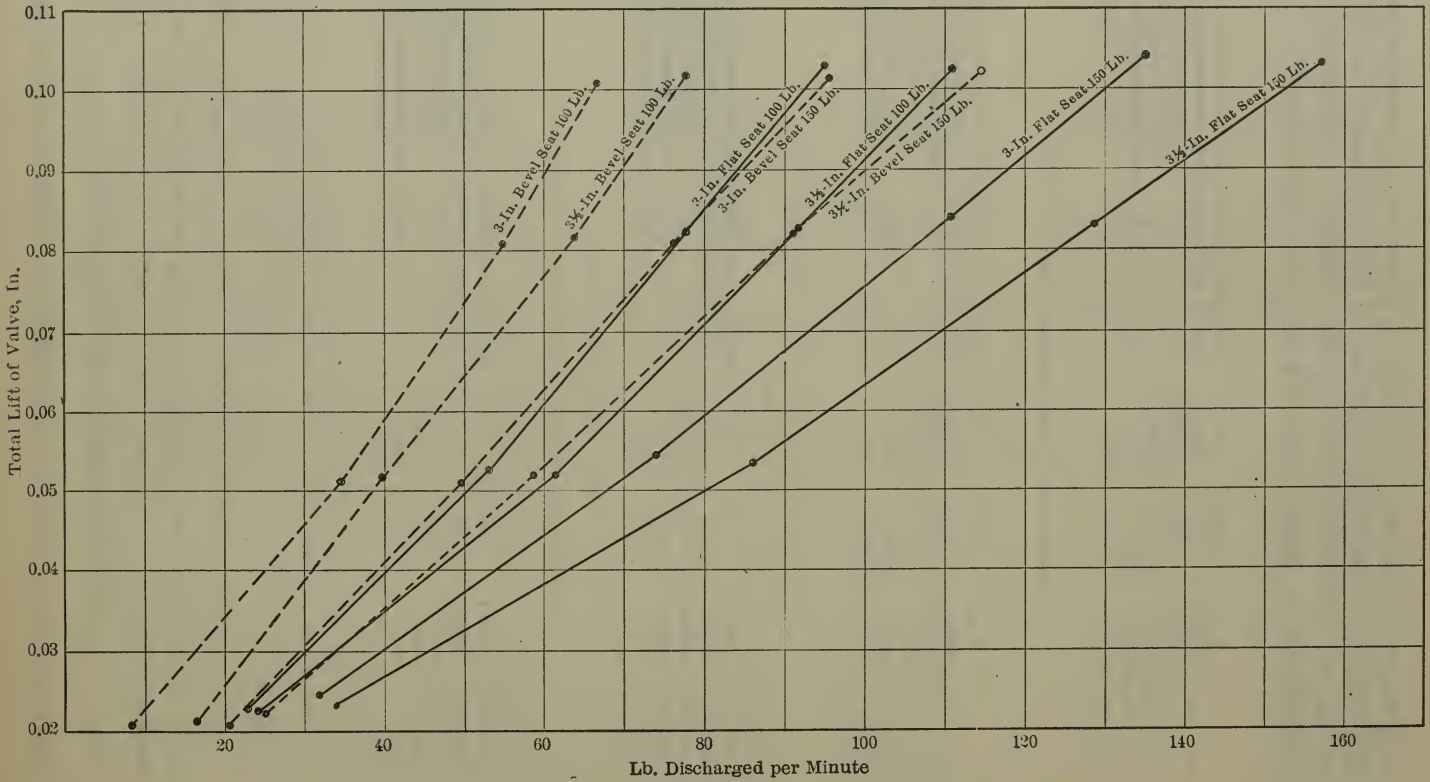


FIG. 7 RESULTS OF TESTS ON CROSBY INSPECTOR POP SAFETY VALVES

ACCURACY OF THE VARIOUS READINGS

11 The lift of the valves could be set with an accuracy as great as that of the 20-thread screw. As the total motion was only $\frac{1}{10}$ in.

TABLE 1 SUMMARY OF RESULTS ON CROSBY MUFFLED LOCOMOTIVE POP SAFETY VALVES

3-In. Flat Seat, Square Edge, Valve marked "J"

Lift of valve as set	0.02	0.05	0.08	0.10
Total lift including yield of metal	0.0247	0.0547	0.0847	0.1047
Lb. discharged per hr.	2845	6313	9342	11222
Boiler pressure gage	208.0	201.0	199.8	201.0
Lb. discharged per hr. reduced to 200 lb. gage	2743	6284	9351	11170
Lb. discharged per min. reduced to 200 lb. gage	45.7	104.7	155.8	186.2

3-In. Flat Seat, Rounded Edge, Valve marked "O"

Lift of valve as set	0.02	0.05	0.08	0.10
Total lift including yield of metal	0.0247	0.0547	0.0847	0.1047
Lb. discharged per hr.	3309	7250	10414	12442
Boiler pressure gage	209.0	205.8	201.3	198.3
Lb. discharged per hr. reduced to 200 lb. gage	3178	7059	10352	12542
Lb. discharged per min. reduced to 200 lb. gage	52.9	117.7	172.5	209.0

3½-In. Flat Seat, Square Edge, Valve marked "A"

Lift of valve as set	0.02	0.05	0.08	0.10
Total lift including yield of metal	0.0251	0.0551	0.0851	0.1051
Lb. disarged per hr.	3688	7688	11319	
Boiler prohessure gage	212.5	204.2	205.0	
Lb. discharged per hr. reduced to 200 lb. gage	3485	7541	11062	
Lb. discharged per min. reduced to 200 lb. gage	58.1	125.7	184.4	

3½-In. Flat Seat, Rounded Edge, Valve marked "B"

Lift of valve as set	0.02	0.05	0.08	
Total lift including yield of metal	0.0251	0.0551	0.0851	
Lb. discharged per hr.	3989	8482	12172	
Boiler pressure gage	209.2	202.7	199.5	
Lb. discharged per hr. reduced to 200 lb. gage	3825	8377	12201	
Lb. discharged per min. reduced to 200 lb. gage	63.7	139.6	203.4	

it is probable that the error from this source is not over half a thousandth of an inch. The difference in expansion due to temperature between the body of the valve and the valve and its spindle was obviated in the case of the locomotive valves by making these

TABLE 2 SUMMARY OF RESULTS ON CROSBY INSPECTOR POP SAFETY VALVE

3-In. Flat Seat

Lift of valve as set	0.02	0.05	0.08	0.10
Lift including yielding of metal	0.0224	0.0524	0.0824	0.1024
Steam discharged per hr., lb	1504	3473	4952	5998
Boiler pressure gage	111.8	109.8	106.3	105.9
Steam discharged per hr. reduced to 100 lb. gage	1364	3200	4694	5705
Steam discharged per min. reduced to 100 lb. gage	22.7	53.3	78.2	95.1
Lift of valve as set	0.02	0.05	0.08	0.10
Lift including yielding of metal	0.0244	0.0544	0.0844	0.1044
Steam discharged per hr., lb	1984	4629	6854	8438
Boiler pressure gage	159.6	157.9	155.9	156.1
Steam discharged per hr. reduced to 150 lb. gage	1875	4415	6617	8137
Steam discharged per min. reduced to 150 lb. gage	31.2	73.6	110.3	135.6

3-In. Regular Bevel Seat

Lift of valve as set	0.02	0.05	0.08	0.10
Lift of valve including yielding of metal	0.0207	0.0507	0.0807	0.1007
Steam discharged per hr., lb	879	2219	3500	4412
Boiler pressure gage	110.4	109.2	108.3	109.8
Steam discharged per hr. reduced to 100 lb. gage	806	2054	3264	4065
Steam discharged per min. reduced to 100 lb. gage	13.4	34.2	54.4	67.8
Lift of valve as set	0.02	0.05	0.08	0.10
Lift including yielding of metal	0.0210	0.0510	0.0810	0.1010
Steam discharged per hr., lb	1313	3049	4670	5788
Boiler pressure gage	161.9	154.9	152.2	151.3
Steam discharged per hr. reduced to 150 lb. gage	1225	2961	4608	5743
Steam discharged per min. reduced to 150 lb. gage	20.4	49.4	76.8	95.7

3½-In. Flat Seat

Lift of valve as set	0.02	0.05	0.08	0.10
Lift of valve including yielding of metal	0.0222	0.0522	0.0822	0.1022
Steam discharged per hr., lb	1479	3636	5408	6760
Boiler pressure gage	105.2	99.2	98.1	102.1
Steam discharged per hr. reduced to 100 lb. gage	1415	3661	5497	6638
Steam discharged per min. reduced to 100 lb. gage	23.6	61.0	91.6	110.6
Lift of valve as set	0.02	0.05	0.08	0.10
Lift including yielding of metal	0.0232	0.0532	0.0832	0.1032
Steam discharged per hr., lb	2070	5236	7772	9286
Boiler pressure gage	153.3	152.8	150.8	147.1
Steam discharged per hr. reduced to 150 lb. gage	2029	5148	7734	9452
Steam discharged per min. reduced to 150 lb. gage	33.8	85.8	128.9	157.5

Table 2—Continued

3½-In. Regular Bevel Seat

Lift of valve as set	0.02	0.05	0.08	0.10
Lift including yielding of metal	0.0215	0.0515	0.0815	0.1015
Steam discharged per hr., lb.	1058	2488	3878	4788
Boiler pressure gage	110.2	105.0	102.0	102.4
Steam discharged per hr. reduced to 100 lb. gage.	972	2384	3810	4690
Steam discharged per min. reduced to 100 lb. gage.	16.2	39.7	63.4	78.2
Lift of valve as set	0.02	0.05	0.08	0.10
Lift including yielding of metal	0.0222	0.0522	0.0822	0.1022
Steam discharged per hr., lb.	1504	3518	5616	6746
Boiler pressure gage	152.2	151.1	154.9	147.4
Steam discharged per hr. reduced to 150 lb. gage.	1484	3495	5454	6854
Steam discharged per min. reduced to 150 lb. gage.	24.7	58.2	90.9	114.2

metals the same. In the case of the flat-seated inspector valves the body and spindle were of cast iron; in the case of the bevel-seated inspector valves the spindle was of composition, but the valve lift was adjusted by the disc-guide threaded below the seat. It will be seen by referring to the drawings that these combinations caused this difference of expansion to be reduced to a minimum.

TABLE 3 3-IN. MUFFLED LOCOMOTIVE POP SAFETY VALVE

Flat Seat with Rounded Edge, Valve Marked O. 0.08 In. Lift as Set, June 16, 1910

Time	Tank, Weight	Weight of Condensed Steam	Average Boiler Pressure (Gage)	Pressure in Muffler (Gage)
10:24	232			
	1120			
10:29		888	204	0
	275			
	1133			
10:34		858	198	0
	261			
	1131			
10:39		870	204	0
	266			
	1155			
10:44		889	206	0
	264			
	1105			
10:49		841	197	0
	265			
10:54	1126	861	199	0

Total time, 30 min.; total steam, 5207 lb.; steam per hour, 10,414 lb.; average boiler gage 201.3 lb.; quality of steam (dry steam =1) =0.996.

12 In testing the valves the adjustments and settings were always made while the valve was hot, practically at the temperature of the valve during a test.

13 The additional lift due to the yielding of the metal because of the presence of steam pressure on the under side of the valve was determined in the case of each valve. The valve was supported by its flange in a testing machine and a load corresponding to 100, 150, or 200 lb. per sq. in. applied to the valve. The movement of the valve with reference to its seat was measured by a micrometer caliper. Different sets of readings on any one valve varied 0.0005 in. and the results are liable to be in error by this amount. It is probable that considering all of the errors in determining the total lift of the valve the result is good to about 0.001 in.

14 The steam was condensed in an Alberger bottom inflow surface condenser with a hot-well at the bottom. This condenser is used in connection with a 500-kw. Parsons turbine in the engineering laboratories at the Massachusetts Institute of Technology. Besides the regular drain to the hot-well there were additional drains of 4-in. pipe, one from each end of the condenser to the hot-well. The level in the hot-well varied about 1 in., which corresponds to about 9 lb. of steam.

15 Each weighing of a tank, empty or full, is good to $\frac{1}{2}$ lb. In the runs with 0.08 and 0.10 in. lifts, if all errors in weighing are assumed to be cumulative and an error of 1 in. in the level in the hot-well be also considered, the maximum error is 15 lb. The probable error is less but even this is a small percentage of the total.

16 In the case of the 0.02 in. lift, the maximum possible error due to all sources would be 10 lb. and for the very worst case this is $\frac{10}{586}$ or less than 2 per cent. The results of the calorimeter tests are good within 2 in the third decimal place.

17 The pressure readings by the boiler gage were taken at 1-minute intervals during most of the time, but occasionally more frequently when the pressure was varying. The gage was tested before and after each series of tests and was found correct.

ADDENDUM BY A. B. CARHART

18 This series of tests of safety valves is notable in that the measurements of discharge are quantitative instead of merely comparative and that the steam is measured by the condensation of the discharge, with determination by calorimeter of the moisture of the

steam entering the valves. Some idea of the large condenser and boiler equipment required may be gained when the total discharge figures are compared with average conditions in ordinary work. And this tremendous rate of steam discharge was not for a few minutes only, but was maintained through 1-hour and ½-hour periods of uniform flow.

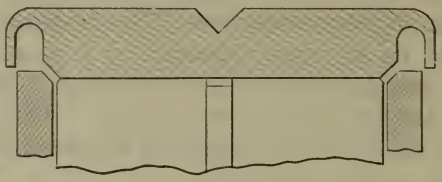


FIG. 8 BEVEL-SEATED VALVE

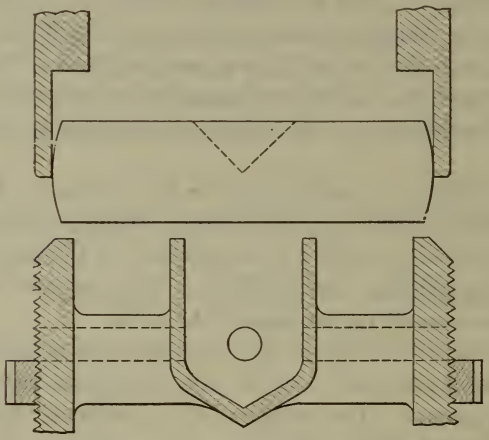


FIG. 9 ANNULAR VALVE

19 Every precaution was observed to avoid errors and all the readings were made by Professor Miller personally. The work was carried on at the Massachusetts Institute of Technology, because no commercial plant was found where the large steam supply could be uniformly maintained under absolute control. A part of the laboratory equipment there was utilized, but it was found necessary to erect a considerable amount of special apparatus and piping, and this preparation for the tests extended over nearly a year.

20 As the valves were adapted from ordinary commercial ar-

ticles, altered in a few details to meet the requirements of convenient orifice regulation as planned by Professor Miller, it may be of importance to note what these peculiarities of construction were, and to supply some of the essential measurements that will be of interest to those who would like to make their own calculations of the areas of the valve passages and orifices.

21 Two of the iron-body valves were of the bevel-seated type, the contacting seat and disc faces forming a single steam-tight joint at an angle of 45 deg. to the vertical. The other two iron-body valves were of the annular type with two concentric seats in the horizontal plane, hence called flat-seated annular valves. The four locomotive valves were all of the flat-seated annular type, but in two of them the inner edge of the larger seat face was made slightly rounded, differing in this detail from the other two similar locomo-

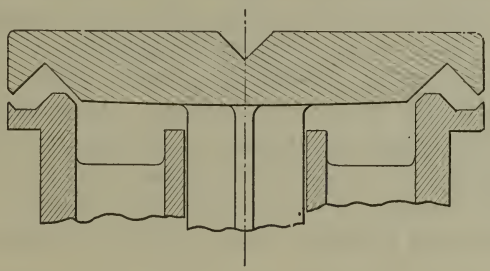


FIG. 10 TYPE OF BEVEL-SEATED VALVE USED IN TESTS

tive valves with which they were to be compared. The effect of this slight change of orifice form and proportion in the locomotive valve seats in increasing greatly the steam discharge is clearly demonstrated.

22 Pop safety valves were invented about 60 years ago, and were perfected in this country about 30 years later and went into general use. Broadly speaking, the invention consists of an addition to the disc of the safety valve, so designed that when the valve is closed such addition is excluded from the action of the steam, but when the valve opens the outflowing steam acts upon it and with the initial force causes the valve disc suddenly to rise higher and the spring to be more compressed than it would be with a force due to the steam pressure upon the original area only; thus there is afforded a prompt,

full-open passage at the seat through which the maximum quantity of steam will flow at once and give relief to the boiler.

23 The bevel-seated valves (Fig. 8) generally have this additional area at the periphery of the disc, outside of the seat, where it forms a chamber with more or less contracted outlet at the extreme lip of the valve, through which all the steam must pass after escaping over the seat before it reaches the open air. As the seat is formed at an angle of 45 deg. to the vertical, the passage between the seat and disc faces when the valve opens is diagonally upward, and what is called the lip of the valve is so related to this seat that the steam is forced to impinge directly upon it before being deflected downward toward the bottom of the chamber where it reacts, the effect of the impact upon the lip of the valve being increased by the expansive force of the steam acting in the lip chamber on account of the partial and momentary confinement before it passes into the open air. This force is regulated in various ways. Sometimes openings are made through the lip or in the floor of the chamber, through which the steam escapes and so reduces the pressure within the lip chamber that the valve disc will not lift too high. Most valves of this kind have a threaded ring surrounding the chamber, providing a micrometer adjustment by means of which the opening at the lip may be made wider or narrower at will. The ring sometimes slightly overlaps the disc at its periphery until the valve has lifted an appreciable amount.

24 Annular valves (Fig. 9) have the additional area located at the center of the disc and within the outer seat. This area is excluded from the action of the steam when the valve is closed, by means of an inner seat, but acted upon directly by the steam when the valve opens. The disc is simply a flat cover for the valve base and has two concentric seats in the same horizontal plane. The central area of the disc, which is not exposed to the steam when the valve is closed, covers a well which communicates to the outer air through four hollow arms radiating from it out through the sides of the valve body. When the valve opens, the steam is discharged directly across the outer flat seat, and there is also a separate additional flow over the inner seat into the central well and out through the four hollow arms. The openings through these arms are controlled by a sleeve, threaded on the valve body, so that it may be turned upward or downward as desired. If it should close the open ends of the hollow arms, preventing the escape of the steam through

the central well, the reactionary effect upon the disc to lift it against the spring would be greater than if the openings from the well were unobstructed; and by locating the sleeve at intermediate points the lift of the valve can be varied.

25 While both styles of valves are efficient, it will be noted that in the bevel-seated type all of the steam passes over the single seat and through the additional regulating chamber at the lip of the valve, and in some cases through a constricted passage at the periphery of the lip; while in the annular valve the steam for the relief of the boiler pressure passes directly through the outer seats without any obstruction whatever, unaffected by any regulation of the by-passed steam through the center that may be necessary. Also, for a given lift of valve, the area of opening through a flat-seated valve is nearly $1\frac{1}{2}$ times that through a valve with bevel seats.

26 In bevel-seated valves the disc is ordinarily guided vertically so that it will return properly to its seat when the valve closes, by means of four or more radiating wings or vanes depending from the face of the disc in the steam space and bearing against the sides of the inlet or throat of the valve (Fig. 8). In the bevel-seated valves tested by Professor Miller, the disc was mushroom-shaped, the guide being small in diameter and acting in an open hub supported in the steam space by arms extending from the sides (Fig. 10). But in the annular valve there are no guides or wings attached to the disc, which is simply a flat member having the truncated form of the central zone of a sphere. This is placed in a cylindrical chamber outside and above the steam space, and there it lifts or moves freely as acted upon by the steam against its lower face and by the opposed spring which presses upon it at its center. The contact between the chamber and the spherical part of the disc is the least possible, which permits universal movement of the disc without possibility of cocking or sticking, and yet insures that the disc will return accurately upon its seats when the valve closes.

27 In the two of the iron-body valves that were fitted with these annular seats, and in two of the locomotive valves, the escaping steam was compelled to make a sharp right-angled turn in passing out over the inner edge of the main discharge seat (Fig. 11). This had been the construction for many years, as the concession to more simple manufacturing operations. It was proposed by Professor Miller to round this corner slightly, as indicated at A' in Fig. 12, at the junction of the side wall of the inlet or throat of the valve with

the main outer seat, for he believed that this would permit a larger flow of steam. Therefore, two other locomotive valves were submitted to test, exactly like the first in all respects except that this edge was rounded, but only to the small extent that the castings would permit without change in patterns. The valves with the angular corner are referred to in the tests as having the "square edge," while the valves with the rounded corner are designated as having the "rounded edge." Fig. 4 shows the later form of these valves with shape of inlet passage modified.

28 The locomotive valves were cast with flanged bases, similar

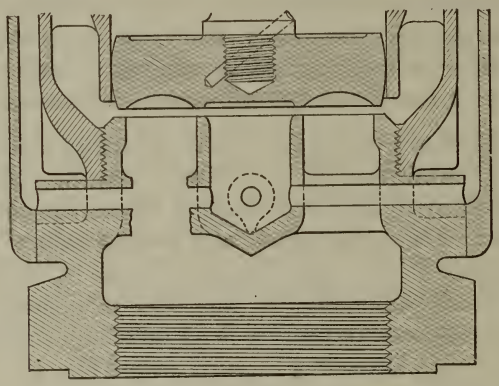


FIG. 11 SECTIONAL VIEW OF ANNULAR VALVE

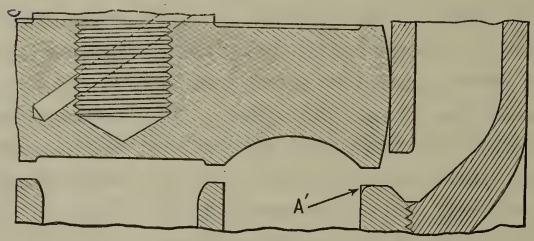


FIG. 12 ANNULAR VALVE SHOWING ROUNDED CORNER OF VALVE SEAT

in size to the iron-body valves, to suit the pipe connections to which they were attached during the tests. The designating letters, *J*, *O*, *A*, and *B*, were stamped upon the locomotive valves before the tests, simply as a convenient means of identification.

29 In both bevel-seated valves, the total width of the seat-face *CBE* (Figs. 13 and 14) was 0.088 in., and the vertical distance *m* between the planes of the lower edge of the seat face at *C* and

the outer edge of the seat face at E measured 0.062 in., in both the 3-in. and $3\frac{1}{2}$-in. valves; therefore when the valve disc was lifted 0.02 in. and 0.05 in., in the tests, the condition was that represented by Fig. 13, and when the disc was lifted 0.08 and 0.10 in., the lower face of the disc at A was above the plane of the outer edge of the seat face at E, as shown by Fig. 14. It will be seen of course that the entrance to the diagonal passage through the seat when the valve is open is measured on the perpendicular line let fall from A to the seat face at B, and this has the mean diameter d, which is greater by the distance $\frac{\text{lift}}{2}$ than the diameter D, which measures the area of the disc that is under pressure when the valve is closed.

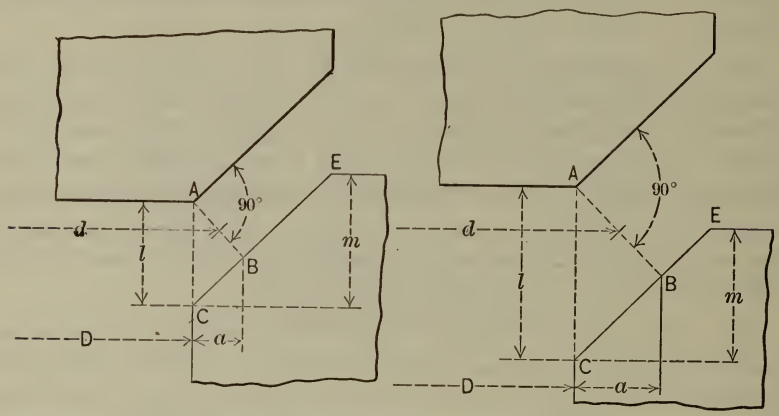
30 It is apparent that calculating the area of the discharge passage upon the greater diameter d instead of the distance D gives some theoretical advantage in this style of valve, yet the total quantity of steam discharged is nevertheless not as great as in the case of the flat-seated valves, in which the free, open passage through the outer seat is measured by the normal circumference of the inner edge of the seat having the diameter D, multiplied by the actual vertical lift of the disc.

31 In the annular valves, the area of the opening into the central well over the inner seat is considerably greater than the outlet area from the well through the four hollow regulating arms, even when these passages are left full-open; for the smaller edge of the inner seat face leading to the central well is 1 1/16 in. in diameter in the 3-in. valve, giving a circumference of 3.33 in., and in the $3\frac{1}{2}$-in. valve the smaller edge of this inner seat over which the steam passes to the central well is $1\frac{1}{4}$ in. in diameter, giving a circumference of 3.92 in.; so that, at 0.08 in. lift for example, the area of the entrance to the well is 0.26 sq. in. in the 3-in. valve and 0.31 sq. in. in the $3\frac{1}{2}$-in. valve. Compared with this, the hollow arms, leading from the central well, are $\frac{1}{4}$ in. in diameter in the 3-in. valve and 9/32 in. in the $3\frac{1}{2}$-in. valve, the four arms giving an additional possible total area of 0.196 sq. in. to be added to the main discharge in the 3-in. valve, and 0.248 sq. in. additional in the $3\frac{1}{2}$-in. valve.

32 Measurements of the pressure in the central well chamber, however, made by connecting a pressure gage to it by means of a separate pipe, show that the pressure existing there when the valve is blowing is appreciably less than the indicated boiler pressure. The

total effective discharge area to be credited to these auxiliary outlets through the central well may therefore be taken as from 4/10 to 8/10 of their nominal measured area just given.

33 In calculating the total effective discharge area of these annular valves, first multiply the smaller circumference of the outer seat by the full vertical lift of the disc and add to this a reduced percentage of the area of the four arms or of the portion of their exit ends not covered by the regulating gate. The effective pressure in these restricted passages is of course considerably reduced, and depends more or less upon the amount of the opening at the seat leading into this central chamber and the freedom of flow into it, for



FIGS. 13 AND 14 DIAGRAMS SHOWING DISCHARGE AREA OF BEVEL-SEATED VALVES

when the disc is lifted from its seat only 0.02 in., there must be an appreciable restriction or wire-drawing of the steam flowing into the central well, and therefore it seems fair to take the value of the discharge area of the hollow arms under these conditions as only 40 per cent of their measured area, increasing to 70 per cent when the lift is 0.05 in. and 80 per cent for the normal lifts of 0.08 and 0.10 in. But these approximations are not important and do not greatly affect the results, for the function of this small by-passed flow of steam is principally to regulate the lift of the disc, and the quantity of discharged steam through the central passages is only incidental and additional, and for all practical purposes these estimated values may be taken as fairly conservative.

34 The width of the seat face on the casting of the base of the

annular valve is approximately 3/16 in. for the outer seat and $\frac{1}{8}$ in. for the inner seat, and the faces on the disc fairly coincide with these when the valve is closed, though the actual width of the contacting faces may be better assumed as about $\frac{1}{8}$ in. for the outer seat and 3/32 in. for the inner seat leading to the central well. The formula for calculating the effective discharge area of the annular valve is:

$$\begin{aligned}\text{Discharge area} &= (\text{circumference of outer seat} \times \text{lift}) \\ &\quad + 4\times (\text{net effective area through each arm}) \\ &= \pi Dl + 0.8 \times \pi(\text{diameter of arm})^2\end{aligned}$$

35 In the bevel-seated valves tested by Professor Miller, the area of the discharge passage over the valve seat may be calculated by the well-known formula, which is made clear by reference to Figs. 13 and 14 illustrating the two cases involved, one where the disc at A does not lift higher than the vertical distance m between the inner and outer edges C and E of the seat face, and the other where the lower face of the disc at A rises above the horizontal plane of the outer edge E of the seat face:

$$AB = l \sin 45 \text{ deg.} = 0.707l$$

$$d = D + \left(2\times\frac{a}{2}\right) = D + \frac{l}{2}$$

$$\text{Seat circumference} = \pi D$$

$$\text{Orifice circumference} = \pi\left(D+\frac{l}{2}\right)$$

$$\begin{aligned}\text{Discharge area} &= (AB)\pi\left(D+\frac{l}{2}\right) \\ &= (0.707l)\pi\left(D+\frac{l}{2}\right) \\ &= 0.707\pi Dl + \frac{0.707\pi l^2}{2} \\ &= 2.22Dl + 1.11l^2\end{aligned}$$

This form of the equation gives results about $1\frac{1}{4}$ per cent larger than the usual approximation

$$\text{Discharge area} = 0.707\pi Dl$$

36 In both types of valves concerned in these tests, the lower end of the inlet connection measures appreciably larger than the actual diameter at the valve seat, and the net area of the unobstructed passage leading up to the edge of the valve seat where the steam escapes is several times greater than the largest possible area through the seat for steam discharge that is intended to be opened in the ordinary operation of the valves. In the case of the annular valves the area

of the inlet passages is approximately four times that of the discharge passages and in the case of the bevel-seated valves it is seven or eight times, assuming in each case that the valve lift is 0.1 in.

37 The accompanying chart, Fig. 15, shows the relative areas in square inches of the narrowest parts of the passages and chambers through which the steam discharged, in the locomotive valves, after passing the valve seat; and the progressive increase explains why the valves were satisfactorily muffled without any retarding effect upon the escaping steam, and why, as noted by Professor Miller, no back pressure could be detected.

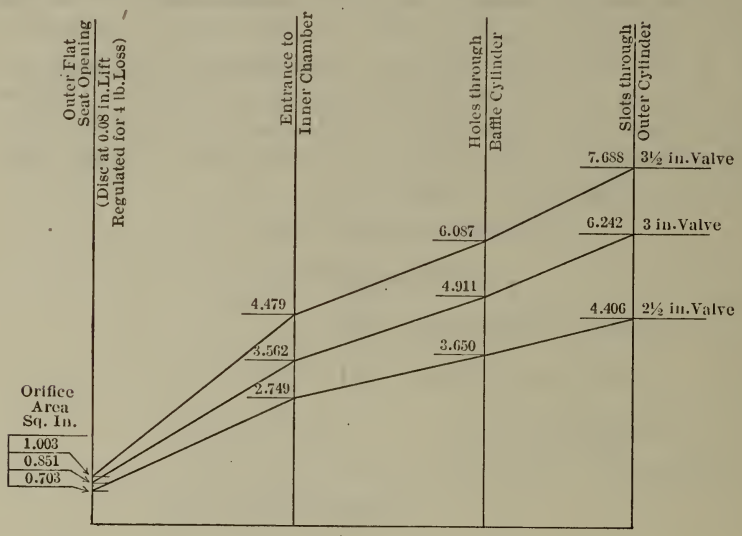


FIG. 15 DIAGRAMS SHOWING PASSAGE AREA AT DIFFERENT POINTS IN THE LOCOMOTIVE VALVES

38 When any safety valve is closed steam-tight, the actual line of contact between the disc face and seat face is a sinuous line without appreciable width, which at points approaches the inner or outer edge of the seat face as happens. Therefore the exposed area of the valve disc, under pressure when the valve is closed, is never exactly the same as the theoretical area calculated by the rated diameter of the valve seat. But for all practical purposes it is nearly so and the diameter of the inner edge of the seat is generally taken as the measure of the disc area.

39 This method of measurement shows a total of 7.07 sq. in.

area of the disc under pressure in the 3-in. bevel-seated valve, when the valve is closed, and 9.62 sq. in. in the $3\frac{1}{2}$-in. valve. In the flat-seated annular valve the net area of the annular face under pressure is 5.79 sq. in. for the 3-in. valve and 7.85 sq. in. for the $3\frac{1}{2}$-in. valve. To obtain these values the areas enclosed within the central seats were deducted. In the 3-in. valve the diameter of the central seat is 1 9/32 in. in diameter and in the $3\frac{1}{2}$-in. valve $1\frac{1}{2}$ in.

40 Multiplying any of these effective areas by the boiler pressure will give the load upon the disc at the moment before the valve opens, which load must be sustained by the valve spring. This is calculated to be of the proper proportions to hold the valve to its seat until the opening pressure is reached and after permitting a full opening of the valve by the lifting of the disc against the spring pressure must return the disc again to its seat and close the valve when the predetermined amount of decrease in boiler pressure has been attained. If the total area of the disc be taken into account, after the valve opens, it is easy to determine the mean pressure of the discharging flow effective to raise the disc, until the reaction of the spring is sufficient to close the valve against the reduced pressure upon the whole face of the valve disc.

41 Assuming, for example, a compression of the spring of 0.50 in. from its free length to exert a load upon the disc equal to the steam pressure of 100 lb. per sq. in., then a lift of the disc of 0.10 in. further would increase the tension of the spring and increase the reactionary load 1/5 more than when the valve is closed. Therefore the discharging steam acting by impact and expansion against the whole area of the disc face when the valve is blowing would have to sustain momentarily a maximum load 20 per cent greater than was exerted by the steam at 100 lb. per sq. in. upon the initial area of the disc when the valve was closed, although after the steam has passed out over the valve seat into the lip chamber or into the central well it is of course at a much lower pressure and has increased considerably in volume.

42 The extreme diameter of the outer edge of the lip of the disc in the 3-in. bevel-seated valve may be taken as $4\frac{1}{8}$ in. when the valve is open, and $4\frac{3}{4}$ in. in the $3\frac{1}{2}$-in. valve. The face of the outer seat of the disc in the annular valve has an appreciable width, making the extreme diameter of the disc acted upon by the steam when the valve is open 3 3/16 in. in the 3-in. valves and $3\frac{3}{4}$ in. in the $3\frac{1}{2}$-in. valves.

43 Although the discharge passage over the outer flat seat of the annular valve is nearly $1\frac{1}{2}$ times as large as is possible in any bevel-seated valve of the same diameter-size and lift, the area of the valve disc exposed to the steam when the valve is closed is, however, only about four-fifths of the total area of the disc, by reason of the central portion within the inner seat circle being shut off from the boiler pressure until the valve opens. This causes less load upon the disc, much less strain and pounding of the valve in seating and less necessary size and spring trouble, than in a bevel-seated valve of the same rated size or seat diameter. When the valve is closed, as the steam pressure of the boiler acts only upon the annulus, or the area between the outer and inner seats, the spring is required only to meet this steam pressure. Therefore the spring has to exert less absolute force than if it held the disc to its seat against the steam acting upon the whole area of the disc. The universal action of the disc due to its spherical edge within the cylinder provides against any binding effect from undue eccentricity of the spring, for all springs in compression necessarily shorten with a torsional movement and side-thrust transmitted to the disc-guides.

44 There is a decided advantage, therefore, in reducing the necessary movement of the spring in its compression and distortion when the valve opens to the minimum amount possible, while yet attaining the required volume of steam discharge.

DISCUSSION

Chas. H. Chase. The records of these tests give an opportunity for the calculation of an average constant for comparison with the constant of Napier's equation and with the constant determined by a series of experiments which were described in a paper [1] presented before the Society in 1909.

That series of tests was made on 3-in., $3\frac{1}{2}$-in., and 4-in. valves of the stationary type, and on $1\frac{1}{2}$-in., 3-in. and $3\frac{1}{2}$-in. locomotive type valves. The valve springs were removed and the lift of the valve disc was controlled in each test by a solid threaded spindle carried by the valve case. The discharge was found by weighing the feed-water, the water level being kept constant. The steam had an average superheat of 37.2 deg. fahr. All of the valves had 45-deg. bevel seats, and the discharge areas were found by the equation

[1] Safety Valve Capacity, P. G. Darling, Trans. Am. Soc. M. E., vol. 31, p. 109.

$$a = 2.22 \times D \times l + 1.11 \times l^2$$

except in two of the 24 tests where the valve lift was greater than the depth of seat and then the equation used was

$$a = 2.22 \times D \times d + 1.11 \times d^2 + D \times (l - d)$$

in which D = diameter of valve, l = valve lift, d = depth of valve seat.

The average constant for the tests was 47.5, to compare with

TABLE 4 VALUES OF CONSTANTS WITH LIFTS AND AREAS OF DISCHARGE

3-In. Regular Bevel Seat
Pressure 100 Lb. Gage

Lift	E	=	Constant	×	a	×	P
0.0207	806	=	50.8	×	0.1383	×	114.7
0.0507	2054	=	52.6	×	0.3405	×	114.7
0.0807	3264	=	52.1	×	0.5447	×	114.7
0.1007	4065	=	52.0	×	0.6819	×	114.7

3-In. Regular Bevel Seat
Pressure 150 Lb. Gage

Lift	E	=	Constant	×	a	×	P
0.021	1225	=	53.0	×	0.1403	×	164.7
0.051	2961	=	52.5	×	0.3425	×	164.7
0.081	4608	=	51.2	×	0.5467	×	164.7
0.1010	5743	=	51.0	×	0.6840	×	164.7

3½-In. Bevel Seat
Pressure 100 Lb. Gage

Lift	E	=	Constant	×	a	×	P
0.0215	972	=	50.6	×	0.1676	×	114.7
0.0515	2384	=	51.5	×	0.4031	×	114.7
0.0815	3810	=	51.9	×	0.6406	×	114.7
0.1015	4690	=	51.2	×	0.8001	×	114.7

3½-In. Bevel Seat
Pressure 150 Lb. Gage

Lift	E	=	Constant	×	a	×	P
0.0222	1484	=	52.1	×	0.1730	×	164.7
0.0522	3495	=	51.9	×	0.4086	×	164.7
0.0822	5454	=	51.2	×	0.6462	×	164.7
0.1022	6854	=	51.6	×	0.8057	×	164.7

the value 51.4 in Napier's equation

$$E = \text{constant} \times a \times P$$

for discharge in pounds per hour.

In the tests of 3-in. and 3½-in. valves with bevel seats now under

discussion, the lift was greater than the depth of valve seat at 0.08 and 0.10 in. However, as a perpendicular from the lower angle of the valve disc does not clear the seat in either case, there remains a discharge orifice bounded by two parallel conical surfaces, as on the lower lifts, and the same method of calculation for discharge area may be used.

Table 4 gives the values of the constants together with the lifts and areas of discharge. Here the average value of the constant is 51.7.

If the calculation of discharge areas for lifts greater than the depth of valve seat is made in the same way as in the tests quoted, the average value of all the constants becomes 49.0. That method is, however, inadmissible for the reason given in Par. 4 of this discussion and we are brought to the conclusion from these tests that for lifts up to 0.10 in. the discharge of dry steam for valves with

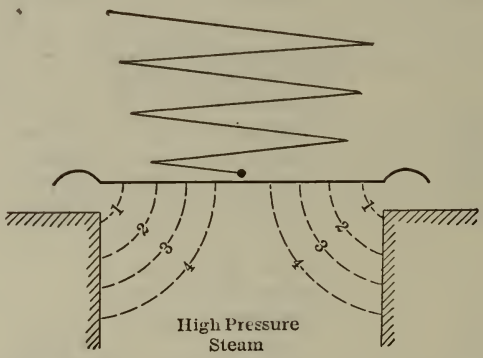


FIG. 16 SHOWING VARYING PRESSURES AFFECTING LIFT OF VALVE

bevel seats may be found very accurately by the direct application of Napier's equation.

W. TRINKS. It occurs to me that the most important part of Professor Miller's paper has been omitted. We can predict with some degree of accuracy how much steam a valve is going to discharge for a given lift, but it takes a very wise man to predict the lift, for this reason: The region around the valve seat is a nozzle so that at line 1, Fig. 16, the pressure will be less than at line 2 and that again less than at line 3, etc.; but the pressure over the whole plate, plus the lip for popping, keeps the valve open against

the compressed spring. For heavy flow there must be equilibrium between spring force and static and dynamic steam forces. To predict the lift of the valve one needs the distribution of the pressure lines 1, 2, 3, 4, etc., over the valve face. While the tests in question were made the distribution of pressure might have been tested. It is desired to know whether such tests were ever made and if so, when they have been published.

A. B. CARHART. The tests were part of the experimental work incidental to proposed improvements in safety valve design, and the measurements obtained were intended to be kept as private information. The purpose was not to determine the maximum amount of steam which could be discharged under any extreme conditions of valve adjustment. The tests give simply the desired positive and quantitative measure of the effect of certain definite modifications of the contour of the orifice at the valve seat, in forms adapted to ordinary commercial uses.

Special care was taken to insure dry steam, and this was shown to be at all times between 0.995 and 0.999 dry, measured close to the safety valve inlet. Obviously the weight of steam apparently passed would have been considerably greater if it had been wet. The steam was measured by condensation of the discharge, for weighing the feedwater would have given only approximations that would not be tolerated in boiler tests or other steam measurements.

The form of the curves, which closely approximate straight lines passing through the origin at zero, indicates well designed proportions of the orifices at the various lifts, and tends to the conclusion that the curves may reasonably be projected further to indicate the discharges for the slightly higher lifts sometimes desired and attained in these valves under actual working conditions.

As the basis of comparison with the results that may be predicted by applying Napier's formula to the calculation of safety valve discharges, the constant derived from Professor Miller's tests might perhaps be more conveniently taken with reference to 70, as used in the ordinary form of Napier's equation, rather than with the transposed value of 51.4 referred to in the form used by Professor Chase. The discharges measured by Professor Miller show a value of the constant of 69.68 as the average of all his measurements of the bevel-seated valves, or a value of 69.93 as the average of the eight tests at the normal lifts of 0.08 and 0.10 in., showing a very close approx-

imation to the value of 70 of Napier's formula. This seems to establish the reasonableness of applying Napier's equation to the calculation of the discharge of safety valves of the type tested by Professor Miller, but the writer thinks these results will hardly justify the general conclusion stated by Professor Chase that "the discharge of dry steam for valves with bevel seats may be found very closely by the direct application of Napier's equation," for obviously the equation can hardly be directly applied to all valves, since in the earlier tests referred to by Professor Chase, with which he makes comparison, the discharges were apparently only about 92½ per cent of the values given by Napier's formula, as Professor Chase shows.

In comparing the present bevel-seated valves with other valves, Professor Chase disregards the fact that although both are called "bevel-seated" (having the contacting faces at an angle of 45 deg. to the vertical), yet the form of the approach to the seat orifice, and the passages and chambers at the lip of the valve were entirely different in design, the extreme lip of the disc of the valves in the earlier tests turning upward, instead of downward as in the valves tested by Professor Miller, and in other ways differing so that they can hardly be directly compared. Moreover, in the earlier tests, no account was taken of the difference in the expansion of the spindle and the valve body, or of the additional lift due to the pressure of the steam against the disc. Professor Miller shows that these may be considerable in amount. The bevel-seated valves tested by Professor Miller were held rigidly in place by screw threads directly below the seat, ending practically on the level with the steam-tight joint, yet even in this case the additional strain beyond the set opening in the various cases was 0.0007, 0.0010, 0.0015 and 0.0022 in., respectively, which gave an additional area of opening, beyond the amount due to the nominal measured lift, averaging 2.9 per cent and in the maximum case amounting to 7.5 per cent increase. In the case of the flat-seated valves, where the control was about 2 in. above the valve seat (Fig. 2) the additional strain due to the steam pressure beyond the measured lift was in the various cases 0.0024, 0.0044, 0.0022 and 0.0032 in. This is several times greater than in the case of the bevel-seated valves, which were controlled much nearer the valve seat; and this greater percentage of increase in discharge area was found, although the area of the disc upon which the additional pressure was exerted

is much smaller than in other valves. If the screw-thread control of the spindle carrying the disc was still further from the seat, at a distance of perhaps 8 or 10 in., the amount of actual opening of the valve beyond the measured opening as set must inevitably be considerably greater than the amounts determined by Professor Miller, and would thus modify the calculation of the percentage of discharge and the relative value of the constant to be compared with Napier's constant in such cases, so as to have reduced appreciably the apparent indicated percentage efficiency of discharge. In Professor Chase's illustration of the effect of using an erroneous formula, which he says is inadmissible, he applies it to 12 cases out of 24 in Professor Miller's report; whereas in the earlier tests he refers to, it was used in only two cases out of 24. Therefore the value of the constant as then determined could not have been greatly affected.

If it were to be assumed, in the case of the flat-seated locomotive valves tested by Professor Miller, that all of the steam discharged had passed over the outer seat alone, the percentage of the discharge would have averaged 108.5 per cent for the valves having the square edge seat and 121.2 per cent for the valves with the seat having the rounded edge, as compared with the discharge to be predicted for the same areas of discharge orifice, calculated according to Napier's formula, taken as 100 per cent. This simply establishes what has been so often asserted, that approximately 15 to 18 per cent of the total discharge of these valves is through the openings through the hollow arms leading from the central well, by which the amount of the valve lift and the closing pressure are regulated, affording therefore this additional by-passed flow of steam through the central passages, independent of and in addition to the main discharge over the outer flat seat. And there is the other interesting comparison, that by reason of these seats being flat, instead of beveled at an angle of 45 deg., the calculated discharge orifice, for the same lift, is nearly 1½ times as great as in the valves of the same seat diameter with the 45 deg. angle.

The original and only purpose of Professor Miller's tests of the locomotive valves was to determine the actual increase in the rate of steam discharge due to rounding the edge of the valve seat. In order to measure exactly the effect due entirely to this change in design, it was necessary to remove the springs and to secure the valve discs at the predetermined lifts. When such

valves are operating in ordinary service with the usual springs the amount of lift is undoubtedly not always the same and is not uniformly maintained throughout the period of blowing; nevertheless these tests do establish the fact that if these valves are

TABLE 5 BEVEL-SEATED, IRON-BODY VALVES AT 100 LB. AND 150 LB. WORKING PRESSURE. AS TESTED BY PROFESSOR MILLER

Size of Valve, In.	Actual Lift	Area of Orifice at Seat Opening	Average Boiler Pressure	Discharge in Lb. per Hour Measured by Tests	Discharge in Lb. per Hour Predicted by Napier's Formula for Given Area and Pressure	Discharge, Percentage, Efficiency upon Napier's Predicted Discharge	Equivalent Constant —Compared with Napier's Value 70
3	0.0207	0.1384	110.4	879	890.43	98.72	70.92
3	0.0507	0.3406	109.2	2219	2170.30	102.24	68.49
3	0.0807	0.5447	108.3	3500	3445.61	101.58	68.96
3	0.1007	0.6820	109.8	4412	4366.75	101.04	69.44
3	0.0210	0.1404	161.9	1313	1275.15	102.97	68.02
3	0.0510	0 3426	154.9	3049	2988.26	102.03	68.49
3	0.0810	0.5468	152.2	4670	4693.42	99.50	70.42
3	0.1010	0.6840	151.3	5788	5839.40	99.12	70.42
3½	0.0215	0.1676	110.2	1058	1076.57	98.28	71.42
3½	0.0515	0.4031	105.0	2488	2481.48	100.26	69.93
3½	0.0815	0.6407	102.0	3878	3845.30	100.85	69.44
3½	0.1015	0.8001	102.4	4788	4818.43	99.37	70.42
3½	0.0222	0.1730	152.2	1504	1484.93	101.28	68.96
3½	0.0522	0.4086	151.1	3518	3484.07	100.97	69.44
3½	0.0822	0.6462	154.9	5616	5636.34	99.64	70.42
3½	0.1022	0.8057	147.4	6746	6716.77	100.43	69.93

Average of 16 tests, percentage efficiency..... 100.517
Average of 8 tests, at normal lifts of 0.08 and 0.10 only 100.191
Average of 16 tests, equivalent value of constant 69.68
Average of 8 tests, at normal lifts of 0.08 and 0.10 only..... 69.93

Orifice areas calculated by full formula, $A = 2.22\ Dl + 1.11\ l^2$.

Data taken from Professor Miller's test of 1910.

Showing rates of discharge slightly exceeding rate predicted by Napier's formula, but fairly confirming its accuracy, as applied to these values.

Napier's formula, W (per second) $=$ Area $\times \frac{\text{Abs. Press.}}{70}$

regulated under the operation of the spring so as to lift 0.05, 0.08 or 0.10 in., as may be desired, then the steam discharge must inevitably be at the rates given in this report. He showed for example that the valves with the rounded seat edge discharged an average of 11.5 per cent more steam than the valves with the

square edge seat, the increase being 12.3 per cent in the case of the 3-in. locomotive valve at 0.05 and 0.10 in. lift.

As it was not Professor Miller's purpose in measuring the discharges of the bevel-seated and flat-seated iron-body valves, to make any direct comparisons with the results of the tests of any other valves or with any other experiments made at different times for other purposes, the writer has not attempted to present any such calculations. But the actual measurements showed an average discharge by the flat-seated iron-body valves 46.5 per cent greater than in the bevel-seated valves at the same fixed lifts; the increase, for example, averaged 52.2 per cent in the case of the 3-in. valves at 100 lb. pressure. The discharge of all the 3-in. flat-seated valves was 49.5 per cent greater than of all the 3-in. bevel-seated valves, and the average increase was 49.2 per cent in the case of all flat-seated valves of both sizes at 100 lb. pressure.

Table 5, showing the comparison of Professor Miller's results with the discharges to be predicted by applying Napier's formula to the nominal discharge areas through the seats of the bevel-seated valves, may be of interest.

E. F. Miller did not desire to present a closure.—EDITOR.

No. 1359

MEETINGS OCTOBER–DECEMBER

LOCAL MEETINGS

PHILADELPHIA, OCTOBER 5

The International Smoke Abatement Exhibition: Paper by George H. Perkins. Published in The Journal for October 1912. An account of the meeting appeared in the issue of November 1912.

NEW YORK, OCTOBER 8

Meeting on Smoke Abatement, using Mr. Perkins' paper. An account appears in The Journal for November 1912.

BOSTON, OCTOBER 16

Water Power Development of the Mississippi River Power Company at Keokuk, Iowa: Paper by Hugh L. Cooper, chief engineer of the company. Contributed by the Boston Society of Civil Engineers.

NEW YORK, NOVEMBER 12

Measuring Efficiency in Manufacturing: Paper by Edward B. Passano, advocating the theory that all profit should be treated as an item of expense. Available in the form of reprints.

NEW HAVEN, NOVEMBER 13

Afternoon Session on Transportation of Materials in Manufacturing Plants: Papers by Herbert L. Seward and by F. C. Bennett of the Stanley Works, New Britain. Evening Session, Industrial Education: Papers by L. P. Breckenridge and F. J. Trinder, principal of the New Haven State Education Shop.

A more complete account of the meeting appears in The Journal for December 1912, and an abstract of Professor Trinder's paper in the issue of May, 1913.

BOSTON, NOVEMBER 15

Dry Rot in Timber used in Slow-Burning Construction: Paper

by Frederick J. Hoxie. An abstract of the paper and discussion appears in The Journal for March 1913.

CHICAGO, NOVEMBER 21

Business meeting for election of new committee, with addresses by engineering societies represented.

ST. LOUIS, DECEMBER 14

Election of local committee, followed by general discussion on engineering subjects of present-day interest.

CHICAGO, DECEMBER 19

Engineering in Panama and South America: Addresses by Col. Henry A. Allen and F. C. Enright, South American representative of the Association of Commerce.

BOSTON, DECEMBER 20

The Testing Laboratory and the Constructing Engineer: Paper by H. W. Hayward, associate professor of applied mechanics of the Massachusetts Institute of Technology. An abstract of the paper and discussion appears in The Journal for March 1913.

THE ANNUAL MEETING

The thirty-third Annual Meeting of the Society, held in New York, December 3-6, 1912, was attended by 561 members and 356 guests. The meeting was noteworthy because of the unusually varied professional program which for the first time was largely under the direction of the sub-committees of the Committee on Meetings. There were nine professional sessions and the last, on Friday, was an all-day session devoted to the discussion of industrial management. The attendance at this session was large, indicating a sustained interest in the proceedings up to the end of the convention.

The various details of the convention which showed much forethought and care in their working out, were in charge of either standing or specially organized committees of the Society. The professional sessions were arranged by the Committee on Meetings, Dr. Charles E. Lucke, Chairman, and by the various subcommittees of this committee, covering various phases of mechanical engineering. The Committee on Meetings in New York, F. H. Colvin, Chairman, coöperated with the House Committee, Francis Blossom, Chairman, in caring for the enter-

tainment features of the meetings, and members of these committees acted as chairmen of the sub-committees, Francis Blossom serving at the head of the Committee on President's Reception, H. R. Cobleigh of the Excursion Committee, and Roy V. Wright of the Acquaintanceship Committee.

A bureau of information was maintained in the foyer of the building during the convention which materially aided in the arrangements for excursions and other details of the meetings.

PROGRAM

Tuesday Evening, December 3

President's Address: The Present Opportunities and Consequent Responsibilities of the Engineer, Alex. C. Humphreys.

Reports of tellers of election of officers and introduction of the President-Elect.

President's Reception in the rooms of the Society.

Wednesday Morning, December 4

Business meeting. Reports of the Council, tellers of election of membership, standing committees. Amendment to Constitution under C 57, relating to election of members and membership grades. New business.

SIMULTANEOUS SESSIONS FOLLOWING BUSINESS MEETING

Power Plant Session

Dimensions of Boiler Chimneys for Crude Oil, C. R. Weymouth.

Discussed by William Kent, E. H. Peabody, D. S. Jacobus.

Tests of a 1000-h.p. 24-Tubes High B. & W. Boiler, B. N. Bump.

Discussed by W. D. Ennis, William Kent, J. H. Browne, D. S. Jacobus. C. D. Young.

Air in Surface Condensation, George A. Orrok.

Discussed by G. J. Foran, A. G. Christie, H. A. Everett, William Weir, J. J. Brown.

Properties of Saturated and Superheated Ammonia, Wm. Earl Mosher.

Physical Properties of Anhydrous Ammonia, L. S. Marks and F. W. Loomis. (Presented through the courtesy of The American Society of Refrigerating Engineers.)

This paper and Mr. Mosher's discussed by A. J. Wood.

Experiments with North Dakota Lignite in a Steam Power Plant and a Gas Producer, Calvin H. Crouch.

Discussion by A. M. Levin.

The Baltimore Sewerage Pump Valve, A. F. Nagle.

No discussion.

Textile Session

The Principles of Valuing Property, Henry K. Rowell.

Discussed by Charles H. Fish, C. J. H. Woodbury, C. T. Plunkett.

The Power Plants of Textile Mills, John A. Stevens.

Discussed by C. T. Main, C. T. Plunkett, C. B. Burleigh.

Gas Power Section

Chairman's Address: Recent Development of Gas Power in Europe, H. J. Freyn.

Discussed by Wm. T. Magruder, D. G. Baker, W. R. Haynie, E. Y. Sayer.

Heavy Oil for Use in Internal-Combustion Engines, Irving C. Allen.

Wednesday Afternoon

Reports of special committees, including Power Tests, Hoisting and Conveying, Standard Cross-Section Symbols, Standardization of Catalogues, Flanges, Pipe Threads.

Reception

Reception to members and guests of the Society, given by the Ladies' Committee in the rooms of the Society.

Wednesday Evening

Reception and subscription dinner tendered to Prof. John E. Sweet, Honorary Member and Past-President, in celebration of his eightieth birthday and in recognition of his services to the engineering profession.

Thursday Morning, December 5

Joint Session in Charge of Sub-Committees on Machine Shop Practice and Iron and Steel

(Papers contributed by Sub-Committee on Machine Shop Practice)

Report of Sub-Committee: The Development of Machine Shop Practice during the Last Decade.

No discussion.

The Vauclain Drill, A. C. Vauclain and Henry V. Wille.

Discussed by J. S. Bancroft, H. P. Fairfield, E. C. Peck.

Efficient Production of Cylindrical Work, C. H. Norton.

Discussed by R. H. Rice, H. P. Fairfield, G. I. Alden, Wm. Kent.

Increase of Bore of High-Speed Wheels by Centrifugal Stress, Sanford A. Moss. (Contributed through the courtesy of the Boston Local Committee.)

Discussed by R. H. Rice, Wm. Kent, W. H. Kenerson, E. A. Sperry.

Investigation of Efficiency of Worm Gearing for Automobile Transmission, Wm. H. Kenerson.

Discussed by Elmer A. Sperry, Wm. Kent, W. C. Marshall, L. B. Taylor.

The Strength of Gear Teeth, Guido H. Marx.

Discussed by L. D. Burlingame, R. E. Flanders, W. S. Huson, W. H. Diefendorf, J. S. Bancroft, Wm. Kent, C. H. Norton, W. C. Marshall, T. M. Phetteplace, E. P. Lesley.

(Paper contributed by Sub-Committee on Iron and Steel)

Case Carbonizing, Marcus T. Lothrop.

Discussed by B. E. Bosler, R. R. Abbott, Albert Sauveur, J. A. Matthews.

Railroad Session

(Papers contributed by Sub-Committee on Railroads)

Train Lighting, H. A. Currie and Benjamin F. Wood.

Discussed by R. M. Dixon, W. F. Bliss, Geo. I. Rockwood, C. D. Young, J. R. Sloan, C. R. Gilman.

Factors in the Selection of Locomotives in Relation to the Economics of Railway Operations, O. S. Beyer, Jr.

Discussed by H. H. Vaughan, Simon Hoffman, C. C. Trump, A. W. Gibbs, C. D. Young, G. R. Henderson, F. F. Gaines, W. F. Keisel, Jr.

Cement Session

(Papers contributed by Sub-Committee on Cement Manufacture)

Some Remarks on the Depreciation Factor in the Cost of Producing Portland Cement, Frederick H. Lewis.

Depreciation and Obsolescence in Portland Cement Plants, H. Struckmann.

Papers discussed together by G. S. Brown, R. K. Meade, R. C. Carpenter, P. H. Wilson, Joseph Brobston, F. W. Kelley, E. Y. Sayer.

The Prevention of Missed Fires in Blasting, W. H. Mason.

The Deterioration and Spontaneous Combustion of Gas Coal, Perry Barker.

Papers discussed together by E. Y. Sayer.

Thursday Afternoon

SESSION ON FLUID FLOW

MEASUREMENT OF AIR IN FAN WORK, Chas. H. Treat.

Discussed by W. H. Carrier, S. A. Moss, J. H. Browne, C. G. deLaval.

THE V-NOTCH WEIR METHOD OF MEASUREMENT, D. Robert Yarnall. (Contributed through the courtesy of the Philadelphia Local Committee.)

Discussed by W. S. Giele, J. H. Browne, F. N. Connet.

THE CENTRIFUGAL BLOWER FOR HIGH PRESSURES, Henry F. Schmidt.

Discussed by R. H. Rice, C. G. deLaval, C. P. Crissey, S. A. Moss, A. E. Guy, W. H. Carrier, C. J. H. Woodbury.

MEASUREMENT OF NATURAL GAS, Thos. R. Weymouth.

Discussed by E. A. Hitchcock, C. J. Bacon, C. C. Thomas.

RECEPTION

Reception to members and guests of the Society, given by the Ladies' Committee in the rooms of the Society.

Thursday Evening

The John Fritz Medal for 1912 was conferred upon Robert Woolston Hunt, Past-President of the Society, " for his contributions to the early development of the Bessemer process." Following the ceremonial there was a joint reception to Mr. Hunt and to the members and ladies of the four American engineering societies.

Friday Morning, December 6

ADMINISTRATION SESSION

Report of Sub-Committee on Administration: THE PRESENT STATE OF THE ART OF INDUSTRIAL MANAGEMENT.

Discussed by C. B. Going, H. P. Gillette, A. H. Church, C. B. Thompson, H. L. Gantt, J. G. Aldrich, R. T. Kent, H. M. Wilcox, F. A. Waldron, F. W. Taylor, F. G. Coburn, David Van Alstyne, Hugo Diemer, C. G. Barth, A. C. Humphreys, H. P. Kendall, Tracy Lyon, D. S. Kimball, H. H. Vaughan, Wm. Kent, R. R. Keely, T. R. Wooley, H. K. Hathaway, S. E. Thompson.

AXIOMS CONCERNING MANUFACTURING COSTS, Henry R. Towne.

Discussed by A. C. Jackson, Augustus Smith, Wm. Kent.

MEASURING EFFICIENCY IN MANUFACTURING, Edward B. Passano.

No discussion.

No. 1360

THE PRESENT OPPORTUNITIES AND CONSEQUENT RESPONSIBILITIES OF THE ENGINEER

By Alexander C. Humphreys, New York

President of the Society

In compliance with a custom now time-honored in The American Society of Mechanical Engineers, it falls to my lot, by reason of the honor you conferred upon me a year ago, to open this 33d Annual Meeting with the President's address.

I have selected as my text: "The Present Opportunities and Consequent Responsibilities of the Engineer."

While speaking from my own experience as an engineer, industrial manager, and president of a college of engineering, supplemented by my experiences for some years past as an officer of this Society, I shall call to my aid the past-presidents of the Society. I shall touch on a number of important moot questions, necessarily briefly, and, although I shall do my utmost to avoid misunderstanding and misinterpretation, I know in advance that in this I cannot be entirely successful. We all have had opportunities to observe the generally inconclusive results from the discussions of papers read before our engineering and other societies due to this failure to secure an agreement of minds. The chances for misunderstanding will be increased because I shall not attempt to impose upon your good nature by reading my paper at length.

Doubtless we are all prepared to agree that special qualifications, natural and acquired, must necessarily involve a commensurate responsibility to employ those qualifications in effective service. For many years past I have been deeply impressed with the particular responsibility of the engineer to the community,

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Presented at the Annual Meeting 1912, of The American Society of Mechanical Engineers.

by reason of his special qualifications. If, through his special qualifications, he is competent to render service to the community at large, that service should not be limited to his particular field of work.

My proposition then is that to-day, by reason of the present complex conditions surrounding the industries, there are offered to us as engineers opportunities for public service which are exceptional; and our responsibilities are the greater because these conditions were in part created by members of our profession. These opportunities are offered to us; if we refuse to accept them, we assume a grave responsibility. The fact that others are more than willing to take the lead in the effort to solve the problems which are threatening our well-being as a nation does not relieve us of responsibility, but rather to the contrary. If we leave it to the lawyers, theoretical economists, and others, to shape the measures of reform, then, just so far as we could have bettered the result, the reproach must rest upon us. As far as possible, the work of reform should be the result of sincere coöperation on the part of all conscientious and competent members of the body politic.

If we consider their addresses collectively, I find that there is little or nothing to be said as to the opportunities and responsibilities of the engineer which has not already been said by our past-presidents.

The education of the engineer is necessarily involved in the question of his responsibility. Some of the authors of these addresses have given expression to decided opinions on this subject of engineering education, and I have no doubt that the authors were not all in entire agreement; and yet, as I read their papers, I find that it is not difficult to harmonize the seeming differences.

Let me recall to your minds these addresses and let me quote briefly from them such parts as are more particularly applicable to my present purpose:

February 16, 1880. "The Field of Mechanical Engineering" by Alexander L. Holley, Chairman of Preliminary Meeting. Five pages only of print, but prophetic as to the future of our Society. He anticipates the late movement to broaden our field of usefulness by the appointment of sub-committees of the Committee on Meetings in the interest of certain "departments of industry." Holley says:

Mechanical Engineering underlies all Engineering. . . . In these leading departments of industry which, at first thought, the public would deem quite outside of mechanical engineering, how completely is this art incorporated with their every detail. . . . The genius that reaches the harmony of perfect construction and perfect beauty (which are interchangeable terms) in Nature's inert materials may not be loftier than that which as perfectly utilizes and governs her wild and capricious forces.

Holley advises the inclusion of several classes of membership and speaks of "the advantages of the association of business men with engineers in these societies." He also recommends the holding of meetings in different parts of the country.

Organization meeting, April 7, 1880, "On the Character of the Association, the Nature of its Work and the Methods by which that Work is to be done." Inaugural Address by Robert H. Thurston, first President.

We have often, and sometimes long, suffered from the effects of legal enactments made by ignorant legislators, influenced by personal motives and the selfish advice of scheming lobbyists. This will be corrected when business men have learned to organize and to act concertedly whenever the business of the country is liable to be affected by legislation. It is their right to be heard fully and patiently, and it is their duty to take such action as will secure for them full consideration. . . .

Ours is the duty to discover facts and to deduce laws bearing upon every application of mechanical science and art in field, workshop, school or household.

Annual Meeting, 1881. "Our Progress in Mechanical Engineering," by Robert H. Thurston, President.

We are everywhere giving up the use of that expensive and perishable material wood, and the weak and brittle minerals, and are substituting for them iron and steel. . . . Machinery speeded up, product increased, labor diminished, cost lessened and machinery given greater automatism and higher efficiency both in making ordinary goods and in adaptation to finer grades. The compressed air engine, the petroleum engine and the gas engine are all just coming forward. . . . The solar motor proposed by Ericsson, the inevitably coming motor of some far-distant epoch has, as yet, made no progress beyond the plans and experiments of the inventor. . . .

Double hulls and transverse bulkheads will make these great vessels safe even against the shock of collision with an iceberg. . . .

The navigation of the air is very possibly on the point of real advancement. . . . Thus we are apparently approaching, though still, perhaps, far from this goal, and we may barely venture to hope that the engineer who is to combine the elements of success, all of which are becoming deter-

mined, will in our own day win the fame that awaits the first successful builder of a flying-machine.

. . . . . .

But the individual must be taught, not simply *permitted* to learn as best he can. Education, directed effectively with the object of giving, in least time and at least cost, a preparation for all the duties coming to the learner, whether in daily toil or social life, is called for.

Annual Meeting, 1882. "The Mechanical Engineer, His Work and His Policy," by Robert H. Thurston, President.

Nature rarely turns a sharp corner in any of her great movements. . . . It is from us, if from any body of men, that the world should expect a complete and thoroughly satisfactory practical solution of the so-called labor problem. . . . The elements of social economy are yet to become known to our people; the most obvious principles of statesmanship are yet to be learned by our legislators, and we have still to look forward to a time when our men of business and our working people shall be fairly and respectfully considered by those who direct public policy. . . .

Such bodies as this must aid our legislative assemblies in developing a *Scheme of Industrial Organization* that shall exhibit highest possible efficiency—one that will prepare the children and youth of the country to enter upon lives of maximum usefulness, and to do the work that may be given them to do with ease and comfort while, at the same time, aiding them to attain health, happiness and content, even if not independence and wealth.

The author speaks for a "common school system of general education, which shall give all young children tuition in the three studies which are the foundation of all education, and which shall be administered under compulsory law."

Annual Meeting, 1883. President E. D. Leavitt did not deliver an address.

Annual Meeting, 1884. An address without title, by John E. Sweet, President. Contrasts and compares Mechanics with Science and Literature.

The educated portion of the world look upon a book not merely as so much paper, and printing and binding, but as the thoughtful work of the author, while the same class almost universally look upon a machine as so much wood and iron, running their minds forward to what it does, and how much it will save, and what the patent it worth, rather than backward to the brain work of its author. . . .

Gauging the value of the thing on the democratic principle of the greatest good to the greatest number, the inventors of agricultural machinery will have few rivals. . . . "May the time come when we shall have a museum in which there shall be gathered the finest specimens of workmanship with the masterpieces of our great engineers. . . . Let us hope that if the high tide of human progress is sweeping on toward a more

useful education, that the day may not be far away when he who knows what to do and how to do it will be regarded as the equal of him who only knows what has been done and who did it.

Annual Meeting, 1885. "The Mechanical Engineer, His Position and Mission," by J. F. Holloway, President.

While it is true that scientific and technical training is, and must ever be, of great advantage to the mechanical engineer, there is yet another source from which, after all, he will derive by far the most benefit, and that is—experience. Not necessarily his own experience, but the experience of others, and of all ages as well. And I know of no other way in which he can be so benefited and aided all through his life.

While none will question the value of the engineer in aiding the progress of the past, all will, I think, agree that at no time in the history of the world was he so important a factor as he is to-day. Need I ask whose triumph has contributed most to the welfare of all the world,—the generals who went over the Alps, or the Engineers who went through them? Mont Cenis and St. Gothard answer.

Annual Meeting, 1886. An address without title by Coleman Sellers, President.

I invite your consideration of a variety of topics which appear to me germane to our organization. . . .

The engineer who counts cost as nothing as compared to the result, who holds himself above the consideration of dollars and cents, has missed his vocation. . . . I am safe in saying that no profession requires a broader education than that of the mechanical engineer. He must be a physicist, a merchant, a lawyer, a chemist, and he should know how to express himself in his mother tongue and be master of the modern languages far enough to have access to the scientific publications of other countries. . . . The engineer must of necessity be a hard student; his school days never end. . . . What will fit him to enter the workshop in better condition than now, will fit him better also for any other walk in life. . . . Education which spoils a man for his work by placing him above manual labor through false pride will continue to do him harm.

The author expresses himself as dissatisfied with the primary schools of the country; they should better prepare for the study to follow.

In our schools we are cramming brains with what taxes the memory to the utmost, but which sends into our workshops boys who are themselves startled to find how little they know as compared to those who, almost ignorant of book-learning, are wise in the knowledge of things about them and skillful in the use of their hands.

The author approves of college sports which train eye and hand and strengthen muscles and develop manhood.

Books, however, are not all repositories of truth. . . . We must teach the money value of knowledge and how to determine it.

He concludes:

The American Society of Mechanical Engineers can do no better work than in devising a good system of education and compelling its introduction into our public schools. Then will those who come after us take command and lead in the industries of the world, and the products of their skill and knowledge find ready sale in all the markets of the world.

This is the man who more than any other influenced the appropriation of the E. A. Stevens endowment to establish the first college to be devoted specifically to the teaching of the science of *mechanical* engineering.

Annual Meeting, 1887. "The Engineer; His Commission and His Achievement," by George H. Babcock, President.

Quoting from Alex. L. Holley's address:

The profession of the mechanical engineer underlies all forms of engineering as well as architecture, manufactures and commerce. . . . His commission is to "subdue the earth." . . . Discovery has followed discovery in a geometrically progressing ratio. . . . There is no good reason to doubt that he will become master of the air, as he is now master of the water, and he will fly from place to place with all the certainty and safety of a bird.

Annual Meeting, 1888. President Horace See, by reason of illness, was unable to deliver his address, but it was published in the Transactions of 1889. Opens with this verse from Cowper:

"Knowledge and wisdom, far from being one,
Have oft-times no connection. Knowledge dwells
In heads replete with thoughts of other men;
Wisdom, in minds attentive to their own.
Knowledge is proud that he has learned so much,
Wisdom is humble that he knows no more."

Some contend that the American youth should occupy a higher position than that of a workman; but what is more humiliating than the spectacle of thousands of our countrymen who have no trade, who daily are compelled to beg for work as common laborers. . . . It seems to me that they (the people) would be better off in this world's goods, much better citizens, and the nation far stronger, if a greater number were reared as skilled and intelligent workmen. . . . It would be a great advantage to our youths if the common school education were modified so as to call on all of the faculties of the mind, and not rely so much on cramming the memory.

Annual Meeting, 1889. Address without title, by Henry R. Towne, President. Three topics:

a The joint excursion of American engineers to Europe the previous summer;
b The contemplated International Exhibition in 1892;
c. Brief resumé of affairs of Society.

We are outgrowing our industrial childhood and are rapidly approaching a point where protection, which has done so much to foster our industries, is no longer needed to the same extent as in the past, a recognition of which fact will, in the near future, enable us to enter in competition for the markets of the world on better terms than we have ever done before.

Quotes extensively from Holley's address in 1880, and then says:

Holley's prediction (as to the growth of the Society) has been fulfilled, but it devolves upon us to carry on the work and if we can, to do even better in the future than in the past.

He strongly recommends the holding of the Society's meetings " in all parts of the country."

Annual meeting, 1890, " The Engineer as a Scholar and Gentleman," by Oberlin Smith, President.

As a matter of fact, the modern Engineer, if he be worthy of the name, must be a scholar as regards many important branches of knowledge. To have become this, he must possess a trained intellect and must have been through a course, whether in college or office or shop, in which he has fulfilled the most important condition of all scholarship, *by learning how to learn.* . . . True scholarship, which consists of a general knowledge of the world at large, combined with such particular knowledge of, and experience in, their chosen vocation, as to make it a success.

This social aspect of professional work, and its great importance in furthering his success in life, not only in the way of pleasure but of profit and reputation, is too often ignored by big-brained young men who are full of scientific zeal, but who have not learned sufficient practical respect for the ways of the world.

Recommends strongly the study of commercial economics, paticularly accountancy. Also, *systematic* physical culture for strength and grace of body.

Annual Meeting, 1891. " The Evolution of American Rolling Mills," by Robert W. Hunt, President. An address of the highest value, historically and technically.

But, as has so often and so fortunately happened, the difficulties and seeming hopelessness of the situation forced a solution of the particular case, and more than that, led to an invention which was destined to revolutionize the rail industry of the country.

He referred to the work of our revered past-president, John Fritz, then chief engineer of the Cambria Iron Works.

If there were no differences of opinion and professional and commercial rivalries, we would be without progress.

Annual Meeting, 1892. "The Steam Engine in Modern Civilization," by Charles H. Loring, President. Another paper of great value, historically and technically.

The Steam Engine may be truly called the material saviour of man. . . . The steam engine is a machine which has been the prolific parent of other machines. . . .

Contemporaneous historians have but scantily drawn attention to the immense influence exerted upon modern history by the steam engine. . . . The development of the steam engine could not progress much, if any, faster than the art of boiler-making.

Annual Meeting, 1893. "The Use of Small Sizes of Anthracite Coal For Generating Steam," by Eckley B. Coxe, President. A study in considerable detail. Admirable for its practical consideration of the questions involved in the testing and valuing of coals.

What the man who pays for the coal wants to know—and he is the one for whom the profession generally works—is how much water can be evaporated for $1, with different coals. . . . I am strongly of the opinion that the time has come when it is better for those engaged in professional investigations to give to the public the result of their investigations before they have reached a final conclusion, as it tends to bring out suggestions from others, and to gather facts which will be of great value in the work in hand.

Annual Meeting, 1894. No President's address, owing to the death of President E. F. C. Davis.

Annual Meeting, 1895. "The Modern Drop Press," by C. E. Billings, President. Briefly historical and then strictly technical. Referring to "the manufacture of the then [1854-1855] celebrated Enfield rifle for the English Government, on the interchangeable system," the author says:

To America is thus due the credit of introducing the interchangeable system in the manufacture of firearms, sewing machines, watches, etc.

Annual Meeting, 1896. "The Progress in the Manufacture of Iron and Steel in America, and the Relation of the Engineer to it," by John Fritz, President.

As a beginning I will make a brief allusion to the Mechanical Engineer, showing his origin and growth, and what he has accomplished in the great

field of metallurgy, and especially in the Bessemer and other important steel-making processes.

The millwright is shown as the progenitor of the mechanical and metallurgical engineer. Referring to the engineers of the iron mills at the time the Bessemer process was introduced, he says:

This was the graduating period for them, and no set of men ever worked more faithfully or earned their diplomas more honestly than these men did. Their diplomas were not made of parchment, but of bright ideas, hard work and energy, coupled with a determination which made failure impossible.

Annual Meeting, 1897. "The Telescope Considered Historically and Practically," by Worcester R. Warner, President. (Followed by an extended discussion by John A. Brashear.) An address of great interest along the lines shown by the title. May well be recommended for reading and re-reading from time to time to those who propose that the mathematics and sciences of the past ages should be "scrapped."

Coming to comparatively modern times, the author reminds us of "that most eminent physicist, Joseph Fraunhofer of Munich. He died in 1826 but is today quoted by the physicists and astronomers of the whole world."

Annual Meeting, 1898. Address without title (but it might well be called "The Engineer"), by Charles Wallace Hunt, President.

Each one of us . . . as it were, looks through a colored glass which gives a personal tint to the scene, colored by the effects of our environment, as well as by our personal temperament. . . .

In order that we may proceed in harmony of thought, we must use words in the same sense. . . .

The new discoveries of materials, of forces, and of laws which now succeed each other so rapidly, make a corresponding increase in the range of the work and the responsibility of the engineer of this present day. . . .

The engineer of the user and the engineer of the maker have widely different duties. . . .

These societies thus become, so to say, the saving banks of our civilization, the repositories and guardians of the results of investigations, experiments and experience that otherwise would have been lost to the world. . . .

It is this great fund of information, principally accumulated during the last century, that we draw upon for material for our text books, our general treaties, and our engineering handbooks.

Man builds to master, to resist or to guide the forces of Nature. If he

has rightly judged the conditions, his work stands as a permanent monument of the fact; but if otherwise, the irresistible laws of Nature will develop the defect and discover the ignorance, incompetence or error to every observer.

Annual Meeting, 1899. "Engineering in the United States Navy. Its Personnel and Matériel," by George W. Melville, President. Largely historical.

Referring to Benjamin F. Isherwood and his work on the vessels employed in our Civil War:

He was accused of building engines which were inordinately heavy, which accusation he has never denied. To the mere office engineer this was true, but he realized what they did not, that these engines had to go into the hands of men who were largely untrained and unfamiliar with machinery. . . . To my mind this was the highest proof of his talent as a sound designer. He had the courage to invite criticism from the book-engineer in order that he might ensure success for the country.

The officers of the Navy designated as "Fighting Engineers."

Annual Meeting, 1900. "Some Landmarks in the History of the Rolling Mill," by Charles H. Morgan, President.

I have often found it to be true that the axioms of my neighbors' trade were the problems of my own, the things by daily contact trite to him, of interest to me. . . . Watt's engine *is* the Hercules, but the rolling mill *is his club*. . . . Those were curious days in the light of to-day; when the appraised value of the West was expressed at Washington in such figurative terms as "interminable desert," "arid plain," "impassable mountains," "not worth a pinch of snuff," "region of savages and wild beasts," "deserts of shifting sands, dust, cactus, and prairie dogs." And so it was, and so it would be now, but for the products of the rolling mill.

Annual Meeting, 1901. "The Early History of Open Hearth Steel Manufacture in the United States," by Samuel T. Wellman, President.

The starting of the works was attended with the usual mishaps which always attend the launching of a new enterprise, especially in the iron business.

This whole paper shows how success is won through costly experimenting and temporary failures. A valuable lesson for the young engineer; and particularly valuable today in connection with the appraisal of properties, which many incorrectly think should be appraised on the basis of original or reproductive cost, less depreciation as derived from some life table in connection with statement of age.

Annual Meeting, 1902. No address made by President Edwin Reynolds.

Annual Meeting, 1903. "The Money Value of Technical Training," by James M. Dodge, President.

This paper caused much pain to some of our friends of the schools and colleges who regard the use of the term "money value" in connection with education as indicating a sordid tendency on the part of those of us who agree with the author that "money value" is one test of efficiency.

> Up to a very recent date, the Apprentice System and that of Independent Delving represented the sole methods of acquiring training. . . . The blending of these two methods developed the earlier mechanical engineers and, will, even in the future, enable those sufficiently gifted by Nature and habit to acquire eminence. The progress of the world, however, calls for a better and more speedy means of producing trained men than could ever be developed by the methods of self-instruction. . . .
>
> We are all prone to take extreme cases of success or failure as the basis of our opinions, and lose sight of the fact that it is the average man whose career shows the true force and direction of the current.

Annual Meeting, 1904. "Some Refinements of Mechanical Science," by Ambrose Swasey, President. Closing paragraph:

> We have recounted some steps of the progress that has been made in the measurement of time, of angles and of length, together with some of the refinements in these measurements, but we are confronted with the fact that notwithstanding all that had been accomplished from centuries past down to the present time, there are, as never before, many imperfections requiring new problems in mechanical science to be worked out for the further enlightenment and welfare of mankind.

Annual Meeting, 1905. "On the Safe-Guarding of Life in Theatres: A Study from the Standpoint of the Engineer," by John R. Freeman, President. A document based upon patient, competent study. A statement of actual conditions found and of proposed remedies.

> In the course of my own studies of the theatre and auditorium problem, I have seen almost everywhere conditions affecting the safety of life that would not be tolerated by the managers of our best industrial works, and all from simple failure to know or to give attention.

Annual Meeting, 1906. "On the Art of Cutting Metals," by Frederick W. Taylor, President. An illustrated book of 248 pages, or 319 pages including the discussion, covering the result of 25 years of work, concurrent work of educated engineers and workingmen, and continuous coöperation of ten great industrial establishments, at an expense of $200,000. A rare example of persistent investigation for the development of truth.

Answering parts of the discussion, Mr. Taylor says:

Our interest in the scientific, or what one might call the intellectual element of the art of cutting metals has been, of course, very great; but after all the true measure of the value of this work should be the actual every day help which will be given by these laws to the superintendents, foremen and mechanics in our machine shops.

Annual Meeting, 1907. "The Mechanical Engineer and the Function of the Engineering Society," by F. R. Hutton, President. An exhaustive presentation of the two subjects covered in the title.

Tredgold's definition of engineering "is silent upon that group of engineers concerned with the liberation, the generation and transmission of forces which are potential and are not realized in nature until in accordance with the natural laws some engineer has caused them to appear."

The author points out that while Tredgold did not include them in the "powers of nature," today there must be included "the forces which are economic or social or psychological in their application" when "human beings become the organs and implements of the factory as a tool of production. . .

The Engineer has therefore become an economic factor as he was not conceived to be in that earlier day.

. . . . . .

Any policy or step which gives occasion rightly to charge a tendency for a national body to localize, is an invasion of opportunity and value.

The author considers at length and favorably the reasons for encouraging local meetings of sections of the Society. Such sections may be either territorially grouped, or by topics and common interests."

Is the privilege of service and of function all on one side or has the Society the right to ask from its members a reciprocal duty to itself? The latter, no doubt.

Annual Meeting, 1908. "The Conservation Idea as applied to The American Society of Mechanical Engineers," by M. L. Holman, President.

The author uses as his text the Conservation Congress held in 1908 at the White House, presided over by the President of the United States and attended by the Governors of States and representatives of engineering and other organizations. An admirable example of the engineer as a citizen employing his special knowledge for the instruction of the community.

During the discussions it became apparent that some effort would be required to keep the conference from political bias. . . .

The President particularly desired the coöperation in the movement of the engineers of the United States and subsequently ascribed to the action of the Engineering Societies the credit of inaugurating the conservation campaign on non-political lines. . . .

As new parties are developed the chances of government by the minority become greater, and with a sufficient number of political parties in the field, revolutions will be the order of the day. . . .

One city for years discharged its sewage into the margin of a lake and took its water supply from the same place. . . .

With us, however, civil and religious liberty seems to include unnecessary exposure to disease. At the conference in Washington, the preventable disease problem was practically overlooked, perhaps from the fact that no trust seems to be operating in that field. . . .

I venture to suggest that we might make progress by ascertaining the secret of German frugality and prosperity rather than by compiling masses of figures to prove what is well known, viz.: that we are wasting the resources of Nature like a true prodigal son. . . .

Which one of you, as householder or engineer, will put up with a poor run of coal in order that posterity may have a good coal? The departments of our government demand the best grade and are not willing to take the "run of the mine."

There is, at present, a strong tendency towards bureaucratic development that is inimical to the successful continuity of our form of government. . . .

Concludes: The Engineering Societies must fall in with the conservation idea and see to it that the returns from the Societies are commensurate with the efforts expended in operating them.

Annual Meeting, 1909. "The Profession of Engineering," by Jesse M. Smith, President. First historical and then largely a consideration of Society activities and the interrelationship of the several engineering societies.

The Engineer capable of being at the head of the larger engineering works must know something of many things, several things well and one thing profoundly.

Concludes: The American Society of Mechanical Engineers has before it a future of usefulness to its members and influence in the profession, which is unlimited. It only requires that we stand by our tradition of increasing the membership with men of high quality as Engineers; that the members maintain enthusiastic devotion to good professional work; that they cooperate with each other in the broadest and most friendly spirit to produce that solidarity of membership and devotion to high ideals, which will compel the world to class the profession of Engineering with the other learned professions.

Annual Meeting, 1910. Address without title, by George Westinghouse, President.

After a brief reference to the affairs of the Society, and particularly to the Joint Meeting at Birmingham, England, with the Institution of Mechanical Engineers, and a brief discussion of "the standardization of mechanical devices," in compliance with suggestions made to him, the author devotes the remainder of his address to an "authoritative statement for the records of the Society" of "my work in developing the airbrake." A valuable record of a great work.

Annual Meeting, 1911. "The Engineer and the Future," by E. D. Meier, President.

A century ago the distinction between the civil and military engineer sufficed, but a few decades ago it became necessary to differentiate in turn the mechanical and electrical engineer, while quite recently upwards of a hundred specialties were enumerated in the attempt to define the activities of the profession, each of which has its recognized experts. . . .

Speaking of men of science: It is the duty, the glorious privilege of the Engineer to receive their discoveries with reverent hands, and apply them to the solution of the practical problems of life. . . .

The Engineer then is a devout believer in natural laws. He knows that they are immutable and permit no exceptions. He needs no Supreme Court to define them as reasonable. . . .

The Engineer is responsible for the vast increase in appliances to meet every demand of that most voracious of living beings, man. The mass of mankind needs to be educated to understand and use them properly. He is in honor bound to supply this education.

As we read these papers continuously, we must be impressed with the wide scope of our professional responsibility, and we, more particularly as mechanical engineers, must be impressed with the point so strongly made in the address of Holley, and referred to by many of his successors, that mechanical engineering underlies all engineering. But here it is well for us to keep in mind that all engineering, while differentiating in great variety as to details, necessarily must be considered comprehensively in the planning of nearly all important undertakings.

The words of our past-presidents serve continually to remind us that the responsible engineer must be a completely qualified expert in some one branch of engineering while possessing a broad grasp on engineering activities as a whole, including the ethical and sociological elements of industrial administration.

Especially for the benefit of students in engineering, I suggest that these addresses be gathered together in a separate volume. A thoughtful reading of these papers should serve to

teach these students that the profession they are preparing to enter is committed to public service along broad lines of usefulness. I further suggest, in future lists of past-presidents that Alexander L. Holley's name be included as the Chairman of the preliminary meeting for organization. We may readily believe that he would have been elected to the office of President if he had not been on the first nominating committee, and if it had not been for his untimely death less than two years after the Society was organized.

As we look over the actual record of progress since the writing of the earlier of these papers, we find that the growth and improvement in material things in many cases has surpassed the prophecies of the writers; but in educational and sociological questions, and particularly as to the relations between capital and labor, so-called, I fear the hopes expressed have not been realized. Here then are the opportunities for the men who, if they will, can speak with authority.

In one other particular we must be strongly impressed by the reading of these papers—the great educational value of the study of the history of engineering. To the engineer-student and the young engineer especially, this study is of great value both specifically and culturally. It is a branch of our educational work more or less neglected in many of our colleges of engineering. Those of us who came more directly in contact with Mr. Conrad Matchoss, the representative of the Munich Museum, during his recent visit to the United States, must have realized that here we have opportunities for progress. And intimately connected with this teaching of the history of engineering is the need for the establishment of at least one comprehensive museum, illustrative of industrial progress, organized somewhat along the lines of the Munich Museum. Let us hope that there may be found, and soon found, those willing and able to father such a scheme. It is a hopeful sign that several of the national engineering societies and other interests are preparing to coöperate for the solving of this problem.

The question of the responsibility of the engineer must carry back to those who are responsible for his education and specific training; including not only the colleges of engineering, but also the graded schools and the employers and others in control of those employed.

With regard to the schools in which our boys are prepared for the engineering college or for a direct entry into business or industrial life, may we not charge that these schools are not as a rule conducted for the benefit of the masses? May we not claim that the public schools, supported by taxation, should be administered primarily for the preparing of the masses for self-support? The keynote in our scheme of public education should be thoroughness, and particularly with regard to the three Rs and the elementary studies in general. The course of instruction in our public schools should be shaped for the benefit of those who are *not* to enter college; that is, for those who, by one cause or another, are forced to become wage-earners at eighteen years of age *or less*. In the scheme, adequate provision should be made for those who do not and cannot remain in school beyond the grammar school period. This change would not work an injustice to those who find themselves in a position to continue their book- and laboratory-study in the colleges. The present scheme tends strongly towards superficiality, which must be harmful to all.

In the case of those who are preparing to enter the college of engineering, I hold that they should not be subjected to a specialized preparatory training. I hold that these boys particularly should receive a sound general training, for they are the ones who will obtain later the advantages of the specialized training. The high-school pupils who are not to enter the college of engineering are the ones more in need of instruction in mechanical drawing, manual training, and the like. If in the allotted time, the boys preparing for the college of engineering could get a sound preparation in the general studies and also get the preliminary practical training, so much the better. But the crowded curriculum of the present day tends to prevent this. *To secure thoroughness our public schools must reduce their requirements or increase their efficiency, or both.* And in this connection it must be realized that all culture is not obtained from so-called cultural studies.

Of late years we hear much to the effect that, by reason of the rapid strides made and being made in engineering science, the courses in our colleges of engineering should be extended from four years to five, six, and even seven years. Included in this scheme of extension is the purpose to devote a considerable

portion of the college course to the general or non-technical studies. If this is wise, then why not use the years of preparation most advantageously on the general studies? In the case of the boy preparing for the college of engineering, why begin to specialize before he enters college? The case of the boy who is to go out from the grammar school or high school to become a wage-earner is quite different. We might, if it is practicable, specialize in his case. And here the coöperative system, the alternating of school and shop, or school and business, demands our thoughtful and sympathetic consideration. This same system, carried on a higher plane, is particularly worthy of our professional attention, especially as now being tried so intelligently under Dean Schneider, of the College of Engineering of the University of Cincinnati. In any case, in the four years engineering course, the general studies should not be slighted and the technical studies should be so presented and taught as to give them the greatest possible cultural value.

For the great majority of students, I cannot believe that it is the part of wisdom to extend the college engineering course beyond the four years which is now the general rule. For these men the age of graduation, on the average, is about twenty-two years. At that age a young man intended for our profession certainly should be prepared to earn a living; continuing, however, to be a student, as he must continue to be to his last active day in his profession, if he aims to fill a position of authority.

If the college course is to be extended by reason of the many more things now to be learned, will five, six, seven, ten, or even twenty years be sufficient? My answer is in the negative, and for two reasons. No matter how many years the student remained in college there would be more to learn of engineering science; and much that the engineer needs to know, the college or university cannot teach. But in four years the college of engineering should be able to teach its students, if they have the natural qualifications and have been soundly prepared, how to learn by themselves, and so how to profit effectively by the teachings of experience. Is there not some measure of disadvantage in keeping an engineering student engaged exclusively on the study of theory, even if the theory is presented in as practical form as is possible in the class-room and laboratory? Is it not true that those who employ young engineers who have had only four

years of college environment complain that too many of them are at first more or less disqualified for practical work by having too high an appreciation of, and too great a reliance upon, their college training? The longer they stay in college, the more apparent will be this partial disqualification.

I know that I shall here be misunderstood to the effect that I am arguing against college training for the engineer. Nothing could be farther from the truth. I am arguing in favor of a proper balance between the teaching of the college and the teaching of the school of experience; that is, a proper balance between theory and practice.

Also, I am not arguing against post-graduate courses for the few who may be specially qualified for advanced study in theory and research. Though here I believe that many engage in college post-graduate study who would do better out in the field of practice. A man is not temperamentally qualified for research simply because he thinks he is. The best graduate school for the great majority of engineer graduates is the school of experience. In many cases even research can be more advantageously pursued outside the environment of the college, in direct contact with the conditions governing practice. This is shown in the many instances of the fruits of outside research carried back into the college and there taught to the undergraduates, giving them the data so disclosed and helping to impress upon them the value of and necessity for experience teaching. Why is it, in the face of these facts continually presented to the college instructors, that certain of them are so ready to claim a monopoly of all educational agencies?

It can be claimed that the store of experience teaching thus carried back to the college can there best be classified and standardized. To a large extent this is true. But here, as in every other such question, there are two sides to be considered. Not always is the schoolman qualified to interpret correctly and comprehensively the data thus placed in his hands. Here is where our engineering societies are doing an important work outside of the colleges, as well as in coöperation with them and their professors as individuals. In the cause of increased efficiency this all speaks for coöperation, and particularly for an adequate appreciation of the value of coöperation between schoolmen and practising engineers. To this end the professors

of engineering should be encouraged to engage in practice for their own benefit and for the benefit of their students. This coöperation can be further developed by direct interchange of ideas and data between professors and their *former students*, as practised in some measure in this country, and to a much greater extent, I understand, in some of the German universities.

Are not many of our troubles with respect to education occasioned by the belief on the part of many educators that all education is to be obtained only within the school and college walls? If they do not believe this, do not many of them minimize the educational value of experience in the business world in contact and competition with one's fellows? Having in mind some of the men with whom we are constantly associating, it seems absurd to have to remind ourselves that the school of experience teaches many things which the school and college cannot teach, and that this school of experience also may exert a strong cultural influence.

Even if some of the schoolmen are inclined to underestimate the educational value of experience teaching, we as engineers must not forget, before the first college of engineering was organized, that the engineer was busy at his many tasks, and the mechanical engineer had produced the steam engine,—that most potent progress-maker of the ages.

While referring to experience teaching and the insufficiency of the unsupported college training for the engineer, permit me to give a word of advice to the juniors of our profession. Too frequently young engineer graduates assume the responsibility of acting as consulting engineers before they have had adequate practical experience. A few make the more serious mistake of assuming this great responsibility immediately after graduation, before they have had any practice. The engineer graduate is not an engineer until he is qualified to deal with the practical conditions of his selected specialty. He must be a commercial engineer in the sense that he should regard the money, for the investment of which he is more or less responsible, as a sacred trust. It is not for him to risk his employer's money in experiments for which he is not qualified by specific experience.

Still more unfortunately some of these young men set up as general consulting engineers. The saving clause is that they are probably as competent in one engineering specialty as another.

No man should attempt the role of consulting engineer until he has had a comprehensive and practical experience in his specialty. By doing so the offender brings discredit not only upon himself but upon our profession as a whole.

Having said this to the young graduate, permit me to say a word as to the responsibility of his employer. The young graduate should not be left without guidance in the school of experience. If the employer is an engineer or is competent to direct in the practical things of his specialty, he should see to it that the employee gets every possible opportunity to learn the practical side of the business and to harmonize the theoretical with the practical. Too often the cadet engineer will be difficult to teach and direct because of his unreadiness to believe that he needs this practical experience. Too often he is unwilling to get down to the long hours, dirt, and drudgery which may be required to win this practical experience. It is the duty of the employer to provide the means for this, and, failing in success after a fair, patient, and intelligent trial, to assist the youngster to a better understanding of what is required of the engineer by giving him his discharge. I know of a number of good men who have been saved by this seemingly severe treatment.

In the treatment of the cadet engineer there rests upon the employer a grave responsibilty. Many a young fellow has been unfairly treated because through ignorance, indifference, or stupidity the one in authority has expected more than could in reason be demanded. Some employers, not technical graduates themselves, exaggerate the immediate results to be obtained from a college training. Others, college graduates themselves, forget how little they knew of practical things when they graduated, and they think they are comparing the ignorance of the cadet with what they knew when they graduated; whereas, really they are comparing with what they know now after ten, twenty, or thirty years of practical experience.

Other cadets have the misfortune to fall under the control of so-called "practical men" who despise "book-learning" and are glad of an opportunity to expose the ignorance of the "college boy." It is the duty of the employer, especially if he is an engineer, to see that the young graduates taken into his employ are, as far as possible, saved from these harmful influences, including the conceit of the graduate himself, which work injury to both employer and employed.

From all that I have said as to the opportunities for improvement, let it not be understood that I am condemning the combination of educational agencies which has produced the men who have done so much for this country through its industries. We must realize that the passage of time has wrought of late great changes, not only as to the science of engineering, but in a marked degree as to the sociologic and economic conditions within which we must practice, if we are to meet our responsibilities to both capital and labor. A system which *permits* men of more than average ability or determination to qualify for our profession should be so improved through experience teaching as to enable us to turn out from our educational mills a larger percentage of first-class product and to reduce the percentage of rejections. Many a young man who might have given a good account of himself, if he had been better handled at some turning point in his career in college or practice, has failed because he had not yet developed the stamina to resist the forces, negative and positive, which were opposing his progress. It is a source of gratification to me that I have saved some from breaking under the strain of injustice in the college and in the school of experience; and it is a source of regret and humiliation to me that in certain cases, as I can now realize, I failed in my duty in this respect.

In arguing for saner and better balanced educational methods to meet the conditions referred to, conditions of the past as well as of the present, I have no sympathy with those who condemn all our educational theories and methods as obsolete and therefore to be consigned to the scrap heap. Some English engineer, whose name I have forgotten, said that where the United States engineers and industrial managers had a distinct advantage over his countrymen was in their readiness to scrap obsolete apparatus. This "scrapping" process can be, like everything else, carried to extremes. We should first determine whether by new combinations the apparently obsolete apparatus or machinery cannot again be made commercially efficient. The inefficient elements only should be replaced. And when it comes to the question of scrapping plant, we must determine whether the interest on both the old and the new does not absorb the anticipated savings. I do not hesitate to affirm that in this country, in our enthusiasm for improvement, we have frequently been guilty of wasteful practice in this regard. This, in a marked degree, ap-

plies to much that has been proposed in connection with the scrapping of our educational agencies.

In connection with discussions of industrial questions and the education of the engineer, we hear much of the opportunities for increased efficiency. There are such opportunities and there always will be. But we also hear much of conservation, frequently from the same sources. Efficiency of methods should include true conservation; the elimination of the false and defective, but certainly the retention of all that has been proved through experience to be true and useful. With Patrick Henry I say, "I know of no way of judging of the future but by the past." Many of our modern reformers would change this to read, "I know of no way of planning for the future but by disregarding the past."

With the conserving of the good and useful there should be the constant effort to render these still more useful. The older a method or practice, provided it has stood successfully the test of years, the more it is to be prized. Yet we are told by some reformers, because a thing is old, and for that reason alone, that it should be scrapped. We are told, because of the marvelous progress made in engineering science in late years, that all the teachings of our schools of engineering should be scrapped. We are told that practically all that was taught in these schools twenty-five years ago is now obsolete. Have all the teaching of languages, logic, history, mathematics, physics, and chemistry of the past been rendered valueless because we have advanced in our knowledge of Nature's laws? Apart from more direct arguments, we may say that one of the most valuable lines of instruction for the study of engineering is the history of the development of engineering science. If it is true that all this of the past is valueless, how was it that the men of the past, taught in the schools and outside of the schools, were able to produce these wonderful advances?

What is the specific function of the college of engineering? Is it to store the brain with facts, or what the teachers believe at the time to be facts, or is it to train and discipline the brains of the students to think with facility and accuracy along the lines of their future vocations? Many of us who, for the time, have forgotten much of this store of facts without being seriously embarrassed by the loss, have our answer to this question. Again

let me say that the school and the college cannot complete the training of the engineer; and that these agencies can only prepare the student to learn his profession in the field of practice. Given the personality, it remains for him to develop in the field of practice the initiative for which the formal teaching and drill of the school and college should have prepared him effectively. Included in this preparation should be some teaching of the business side of engineering practice, including the principles of such subjects as accountancy, depreciation, commercial law, patent law, specifications, contracts, and analysis of data.

Whatever may be said as to the room for improvements in the agencies, processes, and methods employed in the education of the engineer, it should need no argument to demonstrate that the leaders in the profession are as a class those best qualified to advise authoritatively in connection with the efforts to solve many of the most serious problems of the day, involved as these are with questions of transportation, public utilities, and the industries in general.. And for the well-being of our country these problems must be solved wisely.

This is an age of reform, and of late years reforms have been so hastily and incompetently put forward as to bring the term reform into reproach. In fact, a significant feature of many of our reform movements is the readiness of the leaders to act on impulse and without anything approaching adequate investigation. As has been well said, it seems to be a characteristic of this class of reformers to go too far and too fast.

It is with deep regret that I express the belief that college professors, especially professors of economics, and even ministers of the gospel have frequently been offenders in this regard. I believe that certain professors are being retained in their chairs who should be displaced. Especially in the case of universities this may be due to a tendency on the part of president and trustees to exaggerate the right of free speech. Certainly a professor in a State university, to say nothing of the endowed universities, should not be permitted to teach doctrines subversive of law and order and the rights of the individual and property under the Constitution. If these men cannot be controlled by those to whom they are immediately responsible, they should be controlled by those who control the universities.

It is to be borne in mind constantly that good intentions do

not qualify one to do expert work, whether that work be in connection with some reform measure or in some other line of public activity. Here is to be found a grave error of which many an honest enthusiast is guilty. The more astute and sordid of the politicians welcome the assistance which they can secure by guiding the enthusiastic and hysterical reformers into lines of activity which will offer opportunities for trading in immunity, obtaining fees for lobbying, or securing remunerative employment as the agents of these enthusiasts.

Righteous indignation is to be welcomed, and we had better start out in our reforms with the conviction that *as long as the world exists* there will be full opportunity for righteous indignation. But righteous indignation does not of itself provide a remedy for the evils at which we rebel; it only supplies the desire, and possibly the purpose, to produce a remedy. Righteous indignation and misdirected zeal may produce conditions practically as bad as those which it was purposed to correct or eliminate. Reform should not depend upon hysterical suggestion and fitful support, but should be actuated and maintained by all the forces suggested by experience; it should be constantly and tenaciously, but not vindictively, pursued, acting along the best lines suggested by competent, sane, impartial investigation.

As one of the forces to be reckoned with, and particularly in the industries in which our profession is so vitally interested and for which we must carry such a burden of responsibility, is the tremendous force of ignorance and prejudice constantly being swept in upon us by the tide of immigration. If we hold that our country is to be a haven for the ignorant and oppressed, it is then our duty to provide for the control of the elements of danger we so accept. If we offer a haven to these people so ignorant of our fundamental laws for the protection of life and property, certainly we have the right and it is our duty to demand that at least they shall be subject to those laws. The laws as they stand must bind them as they bind us until amended according to law.

The record of recent happenings shows us that there are very real dangers to be apprehended from this source. We need only to read the reports of the trial at Indianapolis of the men of influence under indictment for dynamite outrages, the record of the Los Angeles dynamite outrage so closely linked with

this Indianapolis trial, the trial recently ended at Lawrence of men of like influence indicted for murder through suggestion, the trial of the man who attempted recently to kill an ex-president of the United States, the trials of members of the police force and gangsters of this city, and many other like events indicating a confidently bold disregard for the rights and the very lives of those opposed to these enemies of law and order, to convince us that popular government is under trial as it never has been before in our history as a people.

As I have before suggested, the danger does not lie alone with the ignorant and the designing; if that were true, the danger might be met with greater confidence. But we have to see that the enthusiasm of honest but misguided reformers has to be reckoned with; and these people are to be found in our schools, colleges, universities, industrial establishments, and counting houses.

Let me quote the words of Schrank who tried recently to assassinate Ex-President Roosevelt:

> The shot in Milwaukee, which created an echo in all parts of the world was not a shot at an Ex-President, not a shot at the candidate of a so-called Progressive party, not a shot to influence the pending election, not a shot to gain for me notoriety; no, it was simply to once and forever establish the fact that any man who hereafter aspires to a third presidential term will do so at the risk of his life. If I cannot defend tradition, if I cannot defend the country in case of war, you may as well send every patriot to prison.

This is high-sounding and has a truly familiar ring. Two of the men on trial at Lawrence made statements which also indicated that they considered themselves champions of the defenceless and apostles of liberty rather than of anarchy. These appeals to unreason have a strong influence upon those who are at once ignorant and highly emotional. Unfortunately, those who cannot be charged with ignorance in the ordinary sense are led astray by these hysterical appeals to unreason because they appear to reflect a worthy motive.

And yet this man Schrank, who considers himself a hero and martyr, and who is able to put his dangerous views thus plainly and strikingly before us, by the unanimous verdict of a number of competent physicians, has been declared to be a homicidal maniac.

With the conditions at present surrounding us, which I have

barely suggested, is it not of the first importance that our public school system should be kept free from teachings which are dangerous to our commonwealth? Should not the system go farther and exercise a positive influence for sane thinking upon the children under its influence during the most impressionable period of their lives? Not only should these pupils be taught directly respect for the rights of others and respect for constitutional government, but, above all, the aims and methods should be so amended that the thought constantly kept in the minds of teachers and pupils shall be that the pupils are being trained to become self-supporting and so self-respecting units of the body politic. This implies that these boys and girls shall be taught, shall have pressed home upon them constantly, directly and indirectly, that labor of the hands as well as of the brain, if conscientiously performed, is honorable. Undoubtedly, there are many schools in the land where high-minded, sane teachers are doing their best so to influence their pupils. Of this my talks with many of them convince me; but I have been at the same time convinced that most of these faithful servants of the people feel that they are working under the disadvantages of a system which too generally fails to dignify the labor required for self-support and to cultivate pride in all work well performed.

A respect for law and order should be inculcated in our public schools by the maintenance of a more intelligent and more exacting discipline. The State has a right to demand and should demand, if a free education is provided for the youth of the country, that those so benefited shall render prompt and exact obedience to those in authority. In many cases politics comes in to fetter those who are in immediate authority, and this to the great injury of the pupil, the teacher, and the community. Many men and women, now parasites upon society or criminals, would be self-respecting members of the community if they had been trained to respect honest labor and at least directed to, and given a fundamental preparation for, some vocation.

Let the public schools be conducted for the benefit of the masses and not for the benefit of the very few. The many can be provided for readily and effectively without sacrificing or even hindering those who are qualified by natural endowment to enter college. If some who now go to college were discouraged from so doing, it would be no loss to them or to the community.

Upon us as members of the profession of engineering, individually and collectively, rests the responsibility for doing our utmost to check and correct the evils which I have thus inadequately brought back to your attention. Collectively, and particularly as members of this Society, we are responsible because some members of our profession have been at fault as the advocates of hasty and ill-considered measures of so-called reform. Reforms, especially in connection with industrial administration and the relations between capital and labor, which might be inaugurated by our Society or its members after full and candid discussion, should be of decided benefit to the community as well as to our profession.

Certainly in the matters connected with public education our profession should be able to advise and direct, for the members of the profession have unsurpassed opportunities to judge of the effectiveness of the work of the schools by the results as evidenced in the product. I am quite aware that some professional educators will not be slow to charge that the engineer is not an expert in the field of education, but to this we may reply that this is not solely an academic question and the results are showing every day that the educators need the coöperation of all who are qualified to assist. And may we not claim that the members of our profession as a class are brought as close to the hearts of the people as are the workers in any other one vocation?

Here is another opportunity for unselfish, open-minded coöperation, which is so much needed in our country today.

Now let us for a moment review the work of our Society during the year just closing. The record will be placed before you in detail, but I wish to meet the question, Are we progressing? Are we, in some fair measure, meeting our responsibilities?

In membership we have made a net increase from 4007 to 4350, not counting the affiliates of the Gas Power Section, which show a net decrease of 8, and the student affiliates, which show an increase of from 635 to 722. It is of interest here to note that since Past-President Jesse M. Smith made his presidential address in 1909, the membership of the four national societies generally considered together, civil, mining, mechanical and electrical, has increased from 19,000 to 22,800. The many other engineering societies, specialized and local, no doubt would bring the combined membership up to not less than 50,000.

Our Society may look for a much larger increase in membership during the coming year, as a result of the earnest and intelligent work of the Committee on Increase of Membership under the able and enthusiastic direction of Past-Manager I. E. Moultrop. If the proposed amendments are approved, the additional grade of Associate-Member should serve to bring into our Society a considerable number of men capable of broadening our sphere of usefulness. Not only should we do our utmost to interest as large a proportion of the membership in our activities, but we should do all that is proper to bring into our Society those who are eligible under any of the several classes of membership and who are willing to aid in our work, remembering that membership in such a society as this means giving as well as receiving. This broadening of scope of membership is, as I have already shown, in accord with the suggestion made by Alex. L. Holley at the preliminary meeting in 1880.

Chiefly to the work of the Committee on Increase of Membership and the Gas Power Section may be credited the $10,000 received this year from initiation fees and $20,000 added to our annual income. This activity has materially helped us financially, so that after meeting our pledge to pay off our land bonds, at the rate of $6000 a year, we have added $4000 to our surplus.

Under the able direction of our Publication Committee and the staff, not only has The Journal reflected great credit upon all concerned, but it has contributed a net surplus of $2000 to our treasury. This can be compared with an annual deficit of about $10,000 earlier in our history.

The excellent showing with regard to our finances in general is in large measure due to the supervision of the Finance Committee under the untiring and devoted direction of the Chairman, R. M. Dixon.

Mention is due for the most efficient management of the United Engineering Library under the committee, librarian and staff. This library is performing a service to the engineering profession which is not equalled in our country and of which the three responsible societies may well be proud.

A Society such as ours, if it is to grow in usefulness and influence, must not depend upon the activities of the officers and committees, no matter how devoted and self-sacrificing. The policy of the Society should be to engage the membership at large

in its activities to the greatest degree possible. With this reflection in mind it is most gratifying to note that we have made decided progress during the year in organizing additional geographical groups, and in making some progress towards the development of a simple scheme of administration therefor which shall provide as large a measure of home rule as is consistent with our constitutional requirements.

Also, during the year most satisfactory progress has been made in organizing sub-committees of the Committee on Meetings under the chairmanship of Dr. C. E. Lucke. There have been suggested forty sub-committees to cover as many phases of mechanical engineering as are found in the industries of the country. Believing it is wiser to make haste slowly, the full number of these committees has not yet been appointed. The following have been appointed and have organized: Textiles, Administration, Cement Manufacture, Iron and Steel, Air Machinery, Railroads, Hoisting and Conveying, Industrial Building, Fire Protection, Machine Shop Practice.

The work of these sub-committees may be expected to extend materially the usefulness and influence of our Society. The other sub-committees as suggested should be organized just as soon as a sufficient number of specialists can be interested and the money can be found.

Primarily, I believe, due to the initiative of our Secretary, Calvin W. Rice, the engineers of America and Great Britain have coöperated to erect a memorial window in Westminster Abbey to Lord Kelvin. Members of our Society participated liberally in this tribute to a great man and a great scientist.

Your Council has accepted an invitation to participate in a meeting of the Society of German Engineers to be held in Leipzig next summer. It is to be hoped that our members will evidence their appreciation of the attractive program, which our hosts have prepared, by attending in large numbers.

It is proposed that in connection with the San Francisco Exposition in 1915 there shall be an International Congress of Engineering. In concert with the Society of Civil Engineers, the Institute of Mining Engineers, the Institute of Electrical Engineers, the Marine Engineers and Naval Architects, your Council has already taken preliminary action favorable to this undertaking.

A number of most important committee reports, covering technical subjects of vital interest, will be presented during our meeting. Special reference may be made to a notable report prepared by Mr. Geo. A. Barrus as a member of the Committee on Power Tests.

To provide for the whole of the rich program prepared by the Committee on Meetings, our professional sessions will have to be divided into a number of sections.

Even this brief and fragmentary reference to the work of the year serves to show that the Society, acting through its officers, standing committees, special committees, and the staff, has been maintaining its reputation for effective service and has been reaching out into other fields of usefulness. It is for the members at large to appreciate the service thus rendered and to be prompt to do their share by accepting the opportunities for service offered by membership on the committees, present and prospective.

Thus at considerable length, and yet inadequately, I have discussed the opportunities and consequent responsibilities of the engineer of today. If we claim, as we should, to be members of a profession, we must accept the responsibilities involved. We cannot claim that our profession is one of the three learned professions, because the ignorance of the past created a limitation in favor of religion, law, and medicine. But we can claim that though much of that which the engineer must have at his command is not to be learned from books, it by no means follows that his education is therefore less "liberal" than that of the minister, lawyer, or physician.

There appears to have been a tendency, not so apparent at present, to deny to the mechanical engineer the professional position more readily conceded to the civil or mining engineer. This seems unreasonable and indefensible when we study the question and are forced to endorse Holley's claim that mechanical engineering underlies all engineering. The reason for this rather intangible discrimination is in part due, I believe, to the fact that so many of the rank and file of our branch of engineering are engaged in working out the details, more or less important, of undertakings which are under the general direction of civil or mining engineers or others not members of our profession. Many mechanical engineers thus become absorbed in

the invention and development of mechanical devices, possibly of vital importance in the general scheme, and so fail to obtain a grasp on the undertaking as a whole.

The question of precedence need not be raised; there is credit enough for all. As engineers we are committed to the doctrine of efficiency. Efficiency must come from coöperation, not from discussions as to precedence and relative dignity.

Watt's steam engine made Cort's rolling mill possible. Cort's rolling mill opened up to Watt's engine a new sphere of usefulness.

The Panama Canal, under the direction of thoroughly capable engineers, was a failure until the bacteriologist, the physician, and the sanitarian made it possible for white men to live in the fever stricken zone. Now, while under the general direction and control of military and civil engineers, the success of the undertaking largely depends upon the mechanical and electrical engineers.

Then, while confidently asserting our claim to membership among the liberal professions, and accepting to the full the responsibilities which are thereby involved, let us be prompt to recognize that the progress of the world, material and ethical, depends upon the unselfish, intelligent, and devoted coöperation in service of all professions and vocations under the leadership of men of vision, intellect, power, and humanity.

No. 1361

DIMENSIONS OF BOILER CHIMNEYS FOR CRUDE OIL

AT SEA LEVEL AND AT VARIOUS ALTITUDES, WITH CORRECTING FACTORS FOR FUEL ECONOMIZERS

By C. R. Weymouth, San Francisco, Cal.

Member of the Society

The necessary heights for chimneys for fuel oil plants are much less than ordinarily supposed when the boilers are operating at rating and considerably greater than usually supposed when there are heavy overloads. A chimney of undue height causes an excessive quantity of air for combustion and permits an excessive load on the boilers, both resulting in a large waste of fuel.

2 When the chimney height is limited to that necessary for economical air supply at the desired rate of load, it will be impossible for the most careless fireman to cause a material loss of economy, either by overload or excess air supply, or abuse of boilers due to overload. In this sense the chimney may become an important and inexpensive means for boiler plant regulation and in many ways a safeguard against careless firing. Such usage is, of course, of greatest effect in plants operating at uniform load.

3 Although the writer had previously intended investigating this subject, it was not until the San Francisco earthquake of April 18, 1906, had considerably reduced the height of most masonry chimneys, that he was led to an active collection of chimney data. Many of the results were apparently contradictory. Certain chimneys reduced to a height of 30 ft. gave the usual boiler capacity, and others, reduced to a height of 75 ft., showed under certain conditions of service a decrease in boiler capacity.

4 If the total or potential draft of a chimney were available

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for the flow of gases through the chimney and equal to the frictional resistance within the chimney, the capacity of the chimney would then be a maximum. On the other hand, if a large percentage of the draft of a chimney were maintained at the base, it is self-evident that but little draft would be available to induce a flow of gases; and the greater the draft power at the base of chimney, the less the capacity of chimney in weight of gases discharged per unit of time.

5 In this respect a chimney of given diameter and height may be likened to a centrifugal fan of given diameter operating at a fixed rate of revolutions. When operating on free air discharge, the capacity of the fan is very large, but when operating against an increasing pressure, the rate of discharge rapidly decreases. No one would think of computing a table showing fan capacities without specifying the size of fan and the working pressure.

6 In the same sense a table of chimney capacities is incomplete which does not designate the working draft at the base of chimney. This is particularly true for oil fuel, because of the rapid increase in draft resistance, as boilers are overloaded. For an assumed set of conditions, a chimney, as with a fan, has a definite capacity, but no overload capacity.

7 Chimneys should therefore be selected, not with respect to the rated horsepower of the connected boilers, but after consideration of the maximum boiler overload and the draft resistance of boilers at overload.

8 While most chimney tables are reduced, for convenience, to a boiler horsepower basis, the correct designation of capacity is the weight of gases discharged per unit of time. It follows that in determining the weight of gases from the boiler horsepower, two important factors enter: the ratio of air for combustion to that chemically required, and the efficiency of boilers at the maximum load.

9 Knowing the weight of gases to be handled, it is comparatively an easy matter to determine from existing formulæ the frictional loss due to a given chimney and flue. There are, however, but few published data indicating the draft resistance of various types of oil-fired boilers under various rates of firing.

10 Practical experience has shown that altitude is an important factor affecting the capacity of boiler chimneys. Certain rules of thumb are now employed, and certain formulæ have been

incorrectly derived. The writer's treatment of this question is also given in this paper.

WEIGHT OF CHIMNEY GASES

11 The total weight of gases per hour to be handled by chimney in a given plant is dependent upon the rated capacity of the boilers, their maximum momentary overload capacity, the necessary percentage of excess air for a given type of boiler and furnace, and the boiler efficiency for the conditions assumed.

12 Tests have indicated the efficiencies of boilers, the temperatures of waste gases, and the minimum air required for combustion. In every-day practice the negligence and ignorance of firemen, the disarrangement of furnace brickwork, the carbonization of burners and the improper adjustment of dampers make it necessary to provide for a quantity of air in excess of the minimum air supply obtainable in tests, depending on the type of boiler, the character of service, skill of firemen, etc.

13 For the best types of water-tube boilers fitted with Peabody furnace and operating at rating or at overloads, the writer has assumed as ample margin for good practice 50 per cent excess over the air chemically required, measured at chimney inlet. In determining draft resistance of boilers, 50 per cent excess air through the setting has been assumed for boilers at rating, and slightly less at 50 per cent overload, as explained below.

14 For poorly designed and operated furnaces, 50 per cent excess air is an insufficient allowance and the chimney proportions given in the tables must be modified by the designer to meet actual conditions.

15 When new, a boiler plant will require a minimum of air for combustion, and usually the weight of gases passing up the chimney will be but a slight amount greater than that entering the furnaces. Allowance must be made for leaky boiler settings, long masonry flues, etc.; but with steel flues air infiltration should be negligible. The writer recalls a test of an old plant having long brick flues, where the rate of flow of gases up the chimney was about double that into the connected boiler furnaces; and, as a result of this leakage of cold air, the chimney temperature was very much lower than that of the gases leaving the boilers.

16 With expert firing it is possible to reduce the air supply nearly to the theoretical minimum, both at rated load and at over-

loads. At rating, the relatively large furnace volume makes it possible for considerable amounts of air to pass through the furnace without entering the zone of combustion; but as the load is increased the flame more completely fills the furnace and, except with a deliberate wastage, a larger portion of the air admitted commingles with the flames and aids combustion.

17 In the computations which follow the writer assumes 14 lb. of air chemically required per lb. of oil and, based on 50 per cent excess, there would be admitted a total of 21 lb. of air per lb. of oil. To this the products of combustion and steam for atomization add slightly in excess of one pound, so that hereafter 22 lb. of chimney gases per lb. of oil will be assumed.

18 To assist in the computation of the weight of chimney gases Table 1 is presented, giving calculated boiler efficiencies based on assumed conditions, as explained in the foot notes of the table.

19 The temperature of the waste gases and the gas analysis are determined more easily than the boiler efficiency and, based on these factors, the table assists in approximating the actual boiler efficiency and correspondingly the rate of oil consumption. Due allowance should be made for reduction of boiler efficiency due to any accumulation of boiler scale, sooty tubes, etc., after a period of use.

DRAFT RESISTANCE OF BOILERS

20 The draft resistance of a boiler depends on type, size, arrangement of heating surface, type and arrangement of superheater, if any, extent of baffling of gases, type of oil furnace, arrangement and area of air admission to boiler and to furnace, the excess air for combustion over chemical requirements and the rate at which the boiler is operated. The draft resistances of water tube boilers are ordinarily recorded in the rear of boiler, inside the setting, and close to the outlet damper. These drafts, therefore, do not include the resistance of damper and flue.

21 If a boiler be fired with wide open ash doors, the regulation of air being from the damper at boiler outlet, the draft resistance of the boiler will be a minimum. In boilers having a series of up and down passes, the draft within the boiler setting at the top of the first pass may entirely disappear and a positive pressure may result. This condition is due to the injector action of the burner pumping air into the furnace, and to the smaller

weight of the heated column of gases inside the boiler, as compared with external column of air of equivalent height.

TABLE 1 MAXIMUM THEORETICAL EFFICIENCY FOR A GIVEN PER CENT EXCESS AIR SUPPLY AND FLUE GAS TEMPERATURE

Based on Stated Assumptions [1]

Assumed Per Cent Excess Air Supply	Assumed Temperature of Flue Gases, Deg. Fahr.										Assumed Per Cent Excess Air Supply
	375	400	425	450	475	500	550	600	700	800	
	Calculated Boiler Efficiency, Per Cent										
0	81.35	80.78	80.20	79.61	79.03	78.44	77.28	76.11	73.77	71.43	0
10	80.62	80.04	79.41	78.78	78.15	77.52	76.25	74.99	72.47	69.94	10
50	77.90	77.08	76.25	75.44	74.62	73.80	72.16	70.52	67.24	63.97	50
100	74.42	73.37	72.32	71.26	70.21	69.16	67.04	64.95	60.72	56.51	100
150	....	....	....	....	65.79	64.51	61.93	59.35	54.20	49.04	150
200	....	....	....	....	....	59.86	56.81	53.77	47.67	41.58	200
250	....	....	....	....	....	....	51.69	48.18	41.15	34.11	250
300	....	....	....	....	....	....	46.58	42.60	34.63	26.65	300

Approximate Relation between Per Cent CO_2 and Excess Air Supply as per Assumptions below

Per cent CO_2	4	5	6	7	8	9	10	11	12	13	14	15	15.9
Per cent Excess Air	283	205	155	120	93.0	72.2	55.5	41.9	30.6	21.0	12.7	5.6	0

[1] Values are conditioned on the following assumptions:

Average temperature of air for combustion entering the boiler 80 deg.; humidity 80%; air per lb. of oil chemically required for complete combustion, 14 lb.; B.t.u. per lb. of oil as fired, 18,500; chemical composition of the oil as follows: carbon, 86%; hydrogen, 11%; sulphur, 0.8%; nitrogen, 0.2%; oxygen, 1.0%; water, 1.0%.

Per cent of excess air stated is measured at boiler outlet and consequently includes leakage through boiler setting.

The loss by radiation has been taken as 3%. This loss varies with the size of boiler, insulation, etc., being less than indicated for larger sizes of boilers and greater for smaller sizes.

An allowance of 2% for undetermined losses has been made. This quantity is subject to considerable variation and may, in exceptionally favorable instances, be as low as 0.5%.

The following constants are taken from Marks and Davis' Steam Tables: Absolute temperature = 459.6 deg. + t deg. Heat of vaporization at atmospheric pressure = 970.4 B.t.u. Specific heat of superheated steam at atmospheric pressure, for the range of temperature from 212 deg. to 700 deg. fahr. = 0.47.

This table shows conservative efficiencies primarily for use in stack calculations.

22 In boilers with brick settings there may be an escape of hot gases from the top of boiler at times, giving rise to a possible loss of economy and also to annoyance of firemen from

escaping gases. Under certain conditions it is therefore necessary to close the air admission doors partially to insure a draft at all points within the boiler setting, tending to increase the total draft resistance otherwise necessary. This effect is more noticeable at loads approximating rating, or slightly below, than at overloads, where the friction of flow of air for combustion into the boiler and furnace tends to prevent greater pressure within the setting than without.

23 The analysis of over 75 tests under various conditions, on several sizes of boilers of the same general type, fails to reveal a very close relationship between the total draft resistance and the rate of load on boiler. Other conditions being equal, it is evident that the draft resistance increases slightly with an increase in size of boiler for all rates of firing. A 14-tube high Babcock and Wilcox boiler has a draft resistance greater than a boiler 9 tubes high. There is a variation of the injector action of the burner with different types, arrangement of burners, arrangement of checkerwork forming the floor of furnace, position of the burner relative to checkerwork, and the form of flame relative to checkerwork.

24 Engineers and operators vary the size and arrangement of air spaces for admission to the furnace, and the length and area of air ducts leading from the air doors to the furnace.

25 The area of air admission doors has a general relationship to boiler horsepower, but changes in area of these doors are irregular between certain sizes of boilers. In a Babcock and Wilcox boiler of 600 h.p. with a certain furnace arrangement, draft resistances at rating have been noted as low as 0.08 in. and as high as 0.2 in., this variation being due mainly to increasing quantities of air supplied for combustion. A change in furnace arrangement for economical firing gave for this same boiler a draft averaging slightly over 0.10 in. in one case and less than 0.05 in. in another case, at about the same rating and efficiency.

26 The writer's data are compiled from a series of tests made especially for the determination of boiler efficiencies and not with a view to determination of stack capacities. A series of curves was plotted and an attempt made to correct any inequalities due to the position of air admission doors, the ratio of air in excess of chemical requirements, etc. Even with these corrections, for reasons outlined above, the plotted points did not fall on smooth curves.

27 Fig. 1 shows curves of average draft resistances on Babcock and Wilcox boilers of several sizes, at various rates of firing, given for minimum observed air supply. The apparent inconsistency is due to the fact that the tests were made by different persons.

28 Fig. 2 shows curves of corrected drafts conforming as nearly as may be to a rate of air supply 50 per cent in excess of chemical requirements, the curves being rationalized with respect to each other for similar furnace settings, differences being due to variation in size of boiler, etc.

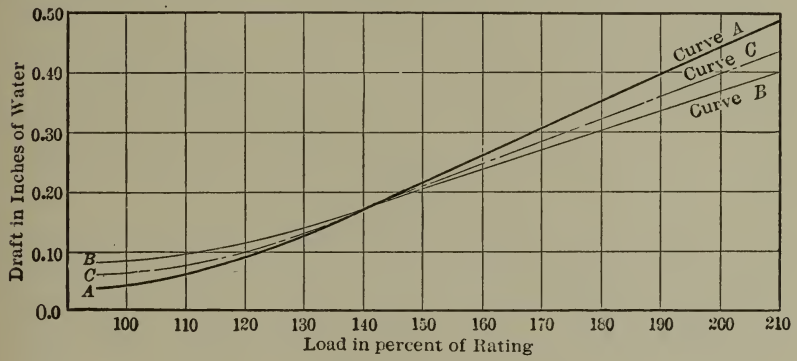

FIG. 1 DRAFT RESISTANCE OF BABCOCK AND WILCOX BOILERS WITH PEABODY FURNACES BURNING CALIFORNIA CRUDE OIL. CURVES PLOTTED THROUGH POINTS OF MINIMUM DRAFT CORRESPONDING TO MINIMUM AIR SUPPLY, TEST CONDITIONS, DRAFT MEASURED IN THIRD PASS.

Curve *A*—21 wide, 14 high, 18 ft. long, 604 h.p.[1]
Curve *B*—21 wide, 12 high, 18 ft. long, 524 h.p.
Curve *C*—14 wide, 9 high, 18 ft. long, 260 h.p.

29 All drafts in Figs. 1 and 2 are measured inside of the boiler between the end of the third pass and boiler outlet.

30 It should be noted that the draft at the rear of boiler will be somewhat dependent on the height above the floor line at the point of measurement. This height varies from about 6 to 20 ft., depending on the type of boiler, and introduces a considerable correction for the chimney height otherwise determined. It is evident that the calculated height of chimney must be measured above the point of observed or calculated draft for the boiler in question, and not from the boiler room floor line.

31 The draft resistance between the third pass and the flue will depend upon the size, type and arrangement of boiler, area

[1]Equipped with double loop superheater.

and position of damper opening, arrangement of circulating tubes, etc.

32 Allowing ordinary coefficients for friction, the calculated draft resistance through the damper box, on the basis of velocity head, does not give a result so great as the average results obtained by experiment. Draft readings are subject to error, owing to the position of draft tube as affected by velocity of

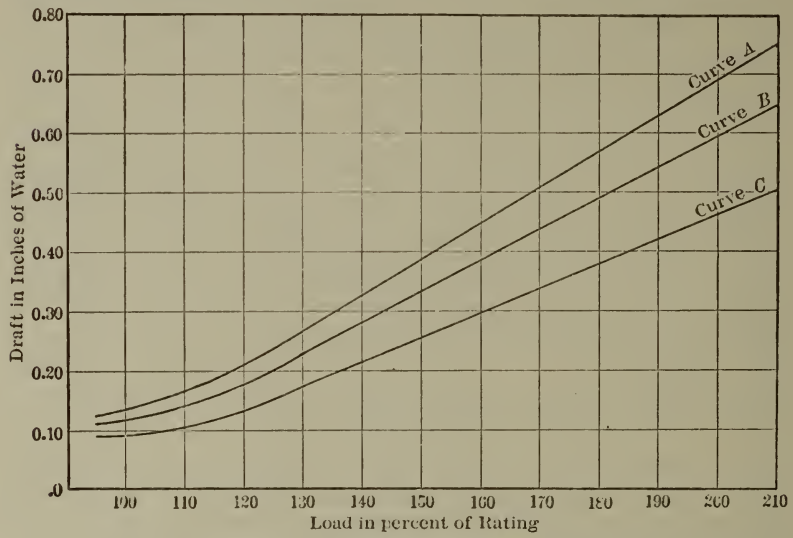

FIG. 2 DRAFT RESISTANCE OF BABCOCK AND WILCOX BOILERS WITH PEABODY FURNACES BURNING CALIFORNIA CRUDE OIL WITH DRAFT MEASURED IN THIRD PASS AND AIR SUPPLY 50 PER CENT IN EXCESS OF CHEMICAL REQUIREMENTS.

Curve *A*—21 wide, 14 high, 18 ft. long, 604 h.p.[1]
Curve *B*—21 wide, 12 high, 18 ft. long, 524 h.p.
Curve *C*—14 wide, 9 high, 18 ft. long, 260 h.p.

gases, etc. In one instance, with a Babcock and Wilcox boiler operating at 100 per cent overload, the draft observations were the same on both sides of the boiler damper, a result manifestly in error. In a 200-h.p. Babcock and Wilcox boiler working at 67 per cent overload, with economical air supply, the draft loss between the third pass and the flue side of damper with damper wide open was 0.04 in.

33 A 520-h.p. Babcock and Wilcox boiler, under good firing conditions, at slightly over rating, gave 0.03 in. loss between flue and third pass, the damper being wide open. Another test on

[1]Equipped with double loop superheater.

boilers of the same size gave 0.02 in. draft loss, at a slight overload, economical air supply.

34 A 680-h.p. Babcock and Wilcox boiler, having overhead flue, outlet between drums, gave the same draft in flue on boiler side of damper as in the third pass, a result no doubt affected by draft differences, due to difference of height of points of measurement.

35 A series of tests on a 600-h.p. Babcock and Wilcox boiler at the Redondo plant 14 tubes high and with a double loop superheater, corrected for 50 per cent excess air supply, gave an average curve from which the following values were taken:

Percentage of Boiler Rating	Draft Resistance, Damper only, In. Water
100	0.08
125	0.11
150	0.14
175	0.16

FRICTIONAL RESISTANCE OF STEEL FLUES, ETC.

36 It is not the purpose of this paper to present new data for frictional resistances, etc., but rather to assume the correctness of established formulæ.

37 The resistance of flues will depend upon the construction, whether of steel, brick, cement, etc., and will also vary with the character of sooty coating, etc. For circular unlined steel flues after a period of use, the writer has assumed the formula

$$D = \frac{3W^2TL}{d^5} \quad [1]$$

in which

D = draft loss, in. of water.
W = weight of gases flowing per second, lb.
T = absolute temperature of gases, deg. fahr.
L = length of flue, ft.
d = diameter of flue, in.

38 The coefficient 3 is merely approximate and although based on information furnished by a prominent fan manufacturer, has been slightly increased to allow for sooty surfaces. The same source indicates that the frictional loss of brick or cement flues would be nearly double that for clean sheet iron, but the writer believes that the loss will not increase in this ratio, owing to the fact that both surfaces will eventually have a coating of soot.

39 Compared with circular flues, the resistance of those of rectangular section is greater at the same velocity by reason of the increased wetted area, being approximately 12 per cent greater for flues of square section and 15 per cent greater for flues of 1 by 1½ rectangular section than for flues of circular section.

40 The resistance of a right angle turn is taken by most authorities as equivalent to the velocity head. Reduced to the basis of the above nomenclature, we have

$$D = \frac{2.6 W^2 T}{d^4} \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

in which

D = draft loss due to velocity head or right angle turn

41 The formulæ for flue resistance are modified for chimneys, as indicated below, the difference in formulæ being, on first thought, not entirely consistent.

42 For resistance of bends of angles greater or less than 90 deg., for diverging or contracting shapes, the designer will be compelled to modify the formulæ on the basis of standard formulæ, the factor of judgment entering largely and the final results being approximate only.

43 The resistance due to a right angle turn in the breeching or at chimney has an important effect on the height of chimney, and to minimize the loss in straight runs as well as in turns, conventional practice requires an area of breeching about 20 per cent in excess of that of chimney.

DRAFT POWER OF CHIMNEY, FRICTION LOSS, ETC.

44 The exact predetermination of the potential draft of a chimney is most difficult, as it is impossible to disassociate the frictional loss within a chimney from the draft loss due to cooling, air leakage, etc.

45 It has been pointed out in Professor Gale's paper[1] that there is a decrease in the potential draft of a chimney, due to friction and cooling of gases within the chimney, to infiltration of cold air, etc., and to atmospheric currents of air in the region of the top of chimney. It has also been shown in Mr. Babcock's discussion of the paper[2] that with a quiet atmosphere there is an

[1] H. B. Gale, Theory and Design of Chimneys, Trans. Am. Soc. M. E., vol. 11, p. 451.

[2] Trans. Am. Soc. M. E., vol. 11, p. 482.

increment of draft due to the column of heated gases directly above the chimney; and, further, that the form of the top of the chimney has a direct bearing on the chimney draft whenever the wind velocity is appreciable.

46 The writer would add that the deceleration of gases within the chimney, due to any cooling in the passage up the chimney, will result in a slight decrease of chimney friction on account of lessening of the volume and velocity of gases; and the conversion of the velocity head into a pressure head will also tend to overcome the downward atmospheric pressure at the top of chimney. It has been suggested to the writer that there are nodal points within the chimney, owing to vibration of the chimney column, and that this phenomenon may interfere with the results calculated from an elementary hypothesis.

47 The effective draft of a chimney will be reduced if ample area-ways are not provided for the inflow of air to the boiler room.

48 From the report of the Liquid Fuel Board, United States Navy, Bureau of Steam Engineering, during 1902 and 1903, the writer noted that in one particular test under forced draft conditions, where the weight of gases passing through the chimney was several times that corresponding to the natural draft conditions, giving rise to a discharge velocity of nearly 60 ft. per second, the calculated chimney friction was greater than 0.5 in. water pressure. The potential draft, due to the weight of chimney gases, was somewhat greater than 0.5 in., so that, neglecting the external influences, the draft at the base of chimney should have been practically negligible. On the other hand, the observed draft was almost as great as the potential draft, indicating a draft increment due to causes other than the difference in weight of the column of air inside and outside of chimney. On other tests the observed draft was considerably reduced, owing in part probably to the brushing aside by the wind of the hot column of gases immediately over the chimney.

49 An examination of the results of very elaborate tests made on a chimney at the Massachusetts Institute of Technology, giving draft observations at various portions, does not indicate a consistent relationship between the observed and calculated draft loss within the chimney for various points along the chimney shaft.

50 Sufficient tests have not been made to determine with accu-

racy the results from all of the influences noted, nor is it necessary that such tests be made. Chimneys must be designed with respect to the unfavorable surrounding conditions and with this idea in view it will be safe to ignore any increment in draft due to the extraordinary influences above cited.

51 Since the existing data in regard to the density of chimney gases refer particularly to coal-burning practice, it is necessary to consider possible variation with oil fuel, which is found to give a density of chimney gases somewhat greater than that with coal firing, varying with the excess air for combustion, the ultimate analysis of the crude oil, humidity of air for combustion, etc.

52 The specific volume of chimney gases is usually defined by

$$S = \frac{T}{K} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

in which

S = cu. ft. of gases per lb.

T = absolute temperature which, according to Marks and Davis = 459.64 + temperature, fahrenheit scale

K = a constant

For dry air, standard barometer, K = 39.78.

53 For chimney gases at a temperature of 500 deg. fahr., 50 per cent excess of dry air being supplied for combustion, K = 39.73. In this instance the value of K is 0.13 of 1 per cent greater for dry air than for chimney gases, indicating a relative lessening in the weight of chimney column to this extent.

54 For 500 deg. fahr. temperature of chimney gases, 100 per cent excess of dry air supplied for combustion, K becomes 39.74, or but slightly heavier than for chimney gases with 50 per cent excess air.

55 For air at 80 per cent humidity, K becomes 39.29, a decrease in density as compared with dry air of 1.2 per cent.

56 For chimney gases at a temperature of 500 deg. fahr., 50 per cent excess air for combustion at 80 per cent humidity, K becomes 39.38.

57 It is thus evident that the relative density of chimney gases is slightly greater than that of air, considering the specified 50 per cent humidity, as against the relative decrease in density when considering dry air only. It follows that with 50 per cent excess air, 500 deg. temperature of chimney gases, there is a de-

gree of saturation of the atmosphere less than 80 per cent humidity, at which the relative density of chimney gases is the same as for air.

58 For 500 deg. fahr. temperature of chimney gases, 100 per cent excess air at 80 per cent humidity, the value of K becomes 39.34, approaching still more nearly than for 50 per cent excess air the relative density of the atmospheric column.

59 In these instances the greatest relative difference in the relative densities of chimney column and air is less than ¼ of 1 per cent, which for ordinary conditions would affect the calculated draft power of the chimney by less than ½ of 1 per cent. Since these differences are small compared with the uncertain value of the correcting factor f, as will be explained, and uncertain chimney calculations in general, it may be considered safe in calculations for chimney draft to ignore the difference in relative density at chimney column and air, assuming throughout the same relative density for both and as equivalent to the atmospheric column based on dry air. A humid atmosphere will affect the performance of a boiler plant. tending slightly towards a sluggish combustion, and by reason of a slightly increased volume and velocity of gases, there is very small increase in the frictional resistances; these effects, however, may be considered of secondary importance.

60 The calculation of the potential draft of chimney is in accordance with

$$D = fKH \qquad [4]$$

in which

f = empirical coefficient depending largely on the rate of cooling gases within the chimney

K = theoretical draft per ft. of chimney due to difference in weight of air and chimney gases, assuming the same relative density for both as for dry air at standard barometer, the chimney column being assumed throughout at the temperature of the entering gases

H = height in ft. of chimney above point of draft measurement

For altitudes other than sea level, the draft varies directly as the normal barometer.

61 Based on the results from actual practice from a number of unlined steel chimneys, the writer has assumed the value $f =$

0.95. This factor is subject to considerable variation. In chimneys having relatively large radiating surface and low velocity of gases, the factor f may be 0.85 or lower. If the chimney be properly designed for a certain maximum load, and all boilers be in operation, the velocity within the chimney will be normal, and the value of f should not vary greatly.

62 A German engineer makes the factor f dependent on the difference between the mean temperature inside the chimney and the mean temperature of external air, but this formula is still at fault in neglecting the chimney velocity and cooling area of chimney surface. It does not appear possible to define the factor f by any simple rational formula.

63 The value of K is determined on the conventional basis. It is important, however, to note that in determining K, when used in connection with the factor f, the temperature should be that of the gases entering the chimney and not the mean temperature of gases within chimney.

64 Table 2 gives the calculated potential draft of a chimney 100 ft. in height for various temperatures of gases entering the chimney and for various temperatures of external air. Column A shows the theoretical draft without loss due to cooling or friction; column B the theoretical draft corrected for a temperature drop in stack and density of flue gases, based on the value $f = 0.95$; column C the net draft based on the stated friction ratio, as referred to below. It is important to note that the drafts are seldom measured at the floor line, and in computing chimney heights, allowance should be made for the height above floor line, of the point of draft measurement.

65 In order to obtain the draft within chimney at the point of entering gases, it is necessary to deduct from the values given in Table 1, Column B, the frictional loss within the chimney, which may be calculated

$$D = \frac{CW^2TH}{d^5} \qquad [5]$$

in which the nomenclature for D, T, W and H will apply as above. C will be represented by a coefficient from practice. For steel chimneys, the writer assumes $C = 2.3$. A German authority gives a frictional coefficient for brick chimneys which reduced to the same form would give $C = 2.5$.

66 The writer has not assumed differing values for steel, brick, and concrete chimneys. While a difference will manifestly

exist, it is believed that the increased friction loss due to masonry will be somewhat offset by the greater rate of cooling of an unlined steel chimney, and with the ordinary heights required for burning oil fuel, these differences will largely disappear.

67 The "Stirling" book recommends that, for brick chimneys, the diameter be made 6 per cent greater than for steel. A German engineering association recommended for steel chimneys as a result of its investigations that the diameter be nearly one-third larger than for brick chimneys.

68 The attendant difficulties in the exact determination of draft loss are so complicated by external influences, as already pointed out, that it is probable that exact information will be long delayed.

69 Based on equation [2] for the resistance of a right angle turn and equation [1] for the resistance of a flue, a simple calculation will show that the loss due to velocity head or the loss due to a right angle turn at flue velocity is equivalent to the resistance of a length of steel flue equal to 10.4 diameters.

70 Based on equation [5] the loss due to velocity head or a right angle turn at chimney velocity will be equivalent to the resistance of a height of steel chimney equal to 13.6 diameters.

TEMPERATURE OF GASES

71 As is well known, the temperature of gases will depend upon the type of boiler, rate of firing boiler, rate of excess air supply, and general condition of boilers as affecting thorough circulation of gases and absorption of heat.

72 In economical types of boilers under test conditions, escaping temperatures are at rating about 425 deg. fahr., or slightly under, and at 50 per cent overload, 500 deg. fahr. or slightly under. With economical boilers under good firing, with tube surfaces in good condition, the boiler temperature should be about 450 deg. fahr. or under, at rating, and about 525 deg. fahr. or under, at 50 per cent overload.

73 It is necessary to allow for a drop in temperature between the boiler outlet and the chimney inlet, depending upon the length and character of the breeching, air infiltration, etc.; also upon the number of idle boilers connected to the same breeching, which may be passing cold air through dampers and which ordinarily are not close fitting.

74 In Table 5 for central chimneys, the writer has

assumed initial temperature of chimney gases 425 deg. fahr. for boilers operating at rating, and 500 deg. fahr. for boilers operating at 50 per cent overload. For chimneys mounted directly over boilers and with short breeching the relative temperature entering the chimney will be greater than indicated. Under unfavorable firing conditions and with less economical types of boilers the temperatures are materially greater than indicated.

ATMOSPHERIC TEMPERATURE

75 As is well known, but frequently overlooked, the temperature of the atmospheric column influences materially the potential draft of a chimney. By referring to Tables 2 and 3 the variation in draft, and correspondingly the variations necessary in chimney height due to range of atmospheric temperature, are appreciable. During cold weather record capacities of chimney are obtained which will not hold for extreme summer heat; the atmospheric temperature may exceed 100 deg. fahr., and in certain locations, 120 deg. fahr. The location of the boiler plant will of necessity decide the consideration given to this factor.

GENERAL FORMULÆ

76 Consideration will here be given to three types of chimneys, differing in respect to the amount of draft lost between the boiler and chimney, as follows:

77 *Case I: Independent chimney for each boiler, mounted directly on boiler, top outlet smoke connection.* In considering this case, the writer has in mind the use of Stirling boilers where an A-shaped breeching is usually employed to connect from the top flue outlet converging to the chimney diameter along lines which involve little frictional resistance within the breeching. In such cases the breeching may ordinarily be considered as the equivalent of the same height of chimney, and a separate loss due to breeching entirely ignored. Where a battery of boilers is connected to one chimney with an A-shaped overhead breeching, the case is but slightly different.

78 *Case II: Independent chimneys direct-connected to a single boiler, or a battery of two boilers, the chimney being located in the rear of boiler or boilers, a short horizontal flue being used.* In this instance the resistance of the breeching itself is almost negligible, but there is added, as compared with Case I, a greater

TABLE 2 CHIMNEY DRAFT—OIL FUEL, SHOWING THEORETICAL AND AVAILABLE DRAFT IN CHIMNEY 100 FT. IN HEIGHT ABOVE THE POINT OF DRAFT MEASUREMENT FOR VARIOUS TEMPERATURES AT SEA LEVEL

Temperature of Gases entering Chimney, Deg. Fahr.	Chimney Draft, In. of Water								
	Temperature of Outside Air = 60 Deg. Fahr.			Temperature of Outside Air = 80 Deg. Fahr.			Temperature of Outside Air = 100 Deg. Fahr.		
	A	*B*	*C*	*A*	*B*	*C*	*A*	*B*	*C*
280	0.436	0.414	0.345	0.382	0.363	0.303	0.332	0.315	0.263
300	0.464	0.441	0.367	0.410	0.390	0.325	0.359	0.341	0.284
320	0.490	0.466	0.388	0.435	0.413	0.344	0.385	0.366	0.305
340	0.514	0.488	0.407	0.460	0.437	0.364	0.409	0.389	0.324
360	0.537	0.511	0.425	0.483	0.459	0.383	0.432	0.411	0.342
380	0.560	0.532	0.444	0.505	0.480	0.400	0.455	0.432	0.360
400	0.581	0.550	0.460	0.527	0.501	0.417	0.476	0.452	0.377
425	0.606	0.576	0.480	0.552	0.524	0.437	0.501	0.476	0.397
450	0.630	0.598	0.499	0.576	0.547	0.456	0.525	0.498	0.416
475	0.652	0.619	0.516	0.598	0.568	0.475	0.547	0.520	0.433
500	0.674	0.640	0.534	0.619	0.588	0.490	0.569	0.541	0.451
525	0.694	0.659	0.550	0.639	0.607	0.506	0.589	0.560	0.466
550	0.713	0.678	0.565	0.659	0.627	0.522	0.608	0.576	0.482
575	0.731	0.694	0.579	0.677	0.643	0.536	0.626	0.595	0.496
600	0.749	0.712	0.593	0.694	0.657	0.550	0.644	0.612	0.510
625	0.765	0.727	0.606	0.711	0.675	0.563	0.660	0.627	0.523
650	0.781	0.742	0.619	0.727	0.691	0.576	0.676	0.642	0.535
675	0.796	0.756	0.630	0.742	0.705	0.588	0.691	0.656	0.547
700	0.811	0.768	0.642	0.756	0.718	0.599	0.706	0.671	0.559
725	0.825	0.783	0.653	0.770	0.731	0.610	0.720	0.684	0.570
750	0.838	0.797	0.664	0.783	0.744	0.620	0.733	0.697	0.581

Column A shows the theoretical draft due to a height of chimney of 100 ft. and $=KH$ where H = height in feet of chimney above point of draft measurement = 100.

K = theoretical draft per foot of chimney due to difference in weight of air and chimney gases at stated temperatures, assuming same relative density for both as for dry air = one one-hundredth of value given in column *A*.

Column B shows draft in *A* corrected for temperature drop in stack and density of flue gases, $B = f A = f KH$. f is variable, but assumed above as 0.95.

Column C shows net draft, being draft in column *B* corrected for friction, assumed as $\frac{5}{6}$ of draft in column *B*.

Chimney friction should be $\frac{1}{6}$ total draft for minimum first cost of chimney when mounted directly over boiler or battery of two boilers. This does not apply to horizontal breeching or to central chimneys combining several losses.

For altitudes, draft varies directly as normal barometer.

TABLE 3 APPROXIMATE MAXIMUM CAPACITY; ACTUAL BOILER HORSEPOWER; STEEL CHIMNEYS; SIZES FOR MINIMUM COST OF CHIMNEY; INDEPENDENT TYPE DIRECTLY MOUNTED OVER SINGLE OR DOUBLE BOILER SETTING WITH TOP OUTLET; 50 PER CENT EXCESS AIR SUPPLY; SEA LEVEL. OIL FUEL.

Temp. of Gases entering Chimney, Deg. Fahr.		300	400	500	600	700	Net Draft required at Point of Measurement In. of Water	Height of Chimney in Ft. above Point of Draft Measurement, for Various Temperatures of Entering Gases and Outside Air, corresponding to Tabulated Draft													
Assumed Boiler Efficiency		80.4	77.1	73.8	70.5	67.3		300 Deg.			400 Deg.			500 Deg.			600 Deg.			700 Deg.		
Lb. of Chimney Gases per B.h.p.		49.5	51.6	54.0	56.3	59.0		Temperature of Air, Deg. Fahr.													
Diam., In.	Area, Sq. Ft.	Actual Boiler Horsepower Temp. Outside Air, 80 Deg. Fahr.						60	80	100	60	80	100	60	80	100	60	80	100	60	80	100
18	1.77	61	62	61	59	56	0.01	2.7	3.1	3.5	2.2	2.4	2.7	1.9	2.0	2.2	1.7	1.8	2.0	1.6	1.7	1.8
21	2.41	90	92	90	87	82	0.02	5.5	6.2	7.0	4.3	4.8	5.3	3.7	4.1	4.4	3.4	3.7	3.9	3.1	3.3	3.6
24	3.14	125	127	125	121	115	0.03	8.2	9.2	10.6	6.5	7.2	8.0	5.6	6.1	6.7	5.1	5.5	5.9	4.7	5.0	5.4
27	3.98	168	172	168	163	154	0.04	10.9	12.3	14.1	8.7	9.6	10.6	7.5	8.2	8.9	6.8	7.3	7.8	6.2	6.7	7.2
30	4.91	219	223	219	212	201	0.05	13.7	15.4	17.6	10.9	12.0	13.3	9.4	10.2	11.1	8.5	9.2	9.8	7.8	8.4	9.0
33	5.94	278	283	278	268	255	0.06	16.3	18.5	21.1	13.0	14.4	15.9	11.2	12.2	13.3	10.1	11.0	11.8	9.4	10.0	10.7
36	7.07	345	352	345	334	317	0.07	19.1	21.6	24.7	15.2	16.8	18.6	13.1	14.3	15.5	11.8	12.8	13.7	10.9	11.7	12.5
39	8.30	422	430	422	408	387	0.08	21.8	24.6	28.1	17.4	19.2	21.2	15.0	16.3	17.8	13.5	14.6	15.7	12.5	13.4	14.3
42	9.62	508	518	508	490	465	0.09	24.6	27.7	31.7	19.5	21.6	23.4	16.8	18.4	20.0	15.2	16.5	17.6	14.0	15.0	16.1
45	11.05	602	615	602	583	553	0.10	27.3	30.8	35.2	21.7	24.0	26.5	18.7	20.4	22.2	16.9	18.3	19.6	15.6	16.7	17.9
48	12.57	708	723	708	685	650	0.20	54.5	61.6	70.4	43.4	48.0	53.0	37.4	40.8	44.4	33.8	36.6	39.2	31.2	33.4	35.8
54	15.90	951	971	951	919	872	0.30	81.9	92.4	106	65.1	72.0	79.5	56.1	61.2	66.6	50.7	54.9	58.8	46.8	50.1	53.7
60	19.64	1240	1260	1240	1197	1134	0.40	109	123	141	86.8	96.0	106	74.8	81.6	88.8	67.6	73.2	78.4	62.4	66.8	71.6
66	23.76	1570	1600	1570	1520	1440	0.50	137	154	176	109	120	133	93.5	102	111	84.5	91.5	98.0	78.0	83.5	89.5
72	28.27	1950	1990	1950	1890	1790	0.60	163	185	211	130	144	159	112	122	133	101	110	118	93.6	100	107

TABLE 3—*Continued*

78	33.18	2390	2430	2390	2310	2190	0.70	191	216	247	152	168	186	131	143	155	118	128	137	109	117	125
84	38.49	2870	2930	2870	2770	2630	0.80	218	246	281	174	192	212	150	163	178	135	146	157	125	134	143
90	44.18	3410	3480	3410	3300	3130	0.90	246	277	317	195	216	234	168	184	200	152	165	176	140	150	161
96	50.27	4010	4090	4010	3870	3670	1.00	273	308	352	217	240	265	187	204	222	169	183	196	156	167	179

Conversion Factors for 60 Deg. and 100 Deg. Atmosphere

Temp. of Gases	300	400	500	600	700
Air at 60 Deg.	1.06	1.05	1.04	1.04	1.04
Air at 100 Deg.	0.935	0.948	0.960	0.965	0.970

This table indicates diameter of chimney for minimum first cost of chimney when cost is proportional to the product of diameter and height. While capacities apply to any arrangement of chimney, cost is minimum only when gases flow vertically into chimney without horizontal breeching and without loss due to right angle turn of gases into chimney. Diameter for minimum cost is independent of draft resistance, if height be selected from the table corresponding to total draft resistance of boiler and damper outlet. For total chimney height above floor line, add to stated heights the actual height above floor line to point of draft measurement. For stated condition of minimum chimney cost, draft resistance is equal to one-sixth of static draft and this allowance has been made in table. See Table 1 for basis of assumed boiler efficiencies, and Table 2 for basis of draft calculations. If boilers have superheaters increase b.h.p. to include heat absorbed by superheater.

draft loss in the damper outlet and turn into flue, also the loss due to a right angle turn of the gases on entering chimney.

79 *Case III: A central chimney connected to a battery of two or more boilers by means of a common flue of appreciable length.* In the computation for this case, the writer will assume the prevailing practice in Babcock and Wilcox installations, there being damper and flue outlet directly in the rear and a straight run of flue to chimney. In this instance there is involved not only the loss through the damper outlet, but the loss due to the turn from the damper outlet into the breeching, loss due to friction of the breeching, and also that due to the right angle turn from the breeching into the chimney. A modification of this arrangement would include installations where there is a resistance due to turns of the breeching, in which instances losses may be minimized by avoiding sharp bends.

80 The fundamental formula, ignoring economizers for the present, is written by equating the potential draft of chimney and the sum of various resistances

$$fKH = D_1 + D_2 + D_3 + D_4 + D_5 + D_6 + D_7 \ldots\ldots [6]$$

in which

f, K, H are as heretofore stated

D_1 = draft resistance of boiler measured on boiler side of damper

D_2 = draft resistance due to damper box

D_3 = draft resistance due to right angle turn into breeching

D_4 = draft resistance due to straight length of breeching

D_5 = draft resistance due to turns in breeching

D_6 = — draft resistance due to turn from breeching into chimney

D_7 = draft resistance of chimney calculated in accordance with [5]

81 D_1 and D_2 usually depend on the boiler in question, and required load or overload. Since the flue area is usually fixed by the chimney area, the values of D_3, D_4, D_5, D_6 and D_7 are interrelated as more fully explained below.

82 When investigating an existing plant under assumed load conditions, it is merely necessary to calculate the various load resistances, comparing their total with that which the chimney is capable of producing.

83 For Case I, D_3, D_4, D_5 and D_6 disappear, and the equation becomes

$$fKH = D_1 + D_2 + \frac{2.3 W^2 HT}{d^5} \quad \ldots\ldots [7]$$

from which

$$\text{B.h.p.} = \frac{60^2}{G} \sqrt{d^5} \sqrt{\frac{fKH - (D_1 + D_2)}{2.3\, HT}} \quad \ldots\ldots [8]$$

in which

G = lb. of chimney gases per b.h.p. per hour

84 Table 3 gives chimney capacities for Case I based on selection of height and diameter for minimum first cost of chimney, as explained below.

85 The treatment for Case II is simple, after considering the more general Case III. For the latter it will be assumed that all flues are rectangular in shape, having a depth of 1.5 times the width, giving a flue resistance 15 per cent in excess of that for a circular flue of equal area and length, which length of flue is ten times the chimney diameter; that flue area is maintained substantially its entire length; and that owing to the decreasing velocities at increasing distances from chimney, the actual resistance of the straight run of breeching will be practically equivalent to that having uniform velocity its entire length, but 5 chimney diameters in length instead of 10 chimney diameters its actual length. D_1 will be as found from test. D_2 and D_3 cannot easily be separated and will be considered as D_2 properly evaluated from test. Then

$$D_4 = \frac{2.3\, W^2 T \left(\frac{5d}{12}\right)}{d^5} \quad \ldots\ldots [9]$$

D_5 disappears from Case III; but when breeching contains a right angle turn, D_5 may be taken as equivalent to the resistance of chimney height of 9.4 chimney diameters. Then

$$D_6 = \frac{2.3\, W^2 T \left(\frac{13.6d}{12}\right)}{d^5} \quad \ldots\ldots [10]$$

Equation [6] may now be written

$$fKH = (D_1 + D_2) + \frac{2.3\, W^2 T}{d^5} \left(H + \frac{18.6d}{12}\right) \quad \ldots\ldots [11]$$

The coefficient $\frac{18.6}{12} = 1.55$; in the table for central chimneys this term, for simplicity of calculation, has been taken as 1.5.

86 As equation [11] involves terms of the fourth and fifth powers, it is simpler to calculate the boiler horsepower of chimney from assumed diameter and height, when [11] finally be comes

$$\text{B.h.p.} = \frac{60^2}{G} \sqrt{d^5} \sqrt{\frac{fKH - (D_1 + D_2)}{2.3\ T\ (H + 1.5d)}} \quad \ldots\ldots\ldots [12]$$

87 In computing the table of capacity for central chimneys, Case III, $D_1 + D_2$ has been taken as 0.15 in. for boilers at rating and 0.3 in. for boilers for 50 per cent overload, which values are conservative for standard boilers, 12 tubes high; for 9 high boilers the height may be slightly reduced, and for 14 high boilers the height should be increased in keeping with draft resistance data above given. Table 4 gives calculated capacities for Case III based on various other assumed data given in tables and explained in the footnotes.

88 For Case II, in which the resistance of straight run of breeching is neglected, the coefficient of d, 1.5, in the term $(H + 1.5d)$, equation [12] becomes instead, $\frac{13.6}{12}$ or 1.13; where one chimney is connected to two boilers symmetrically arranged, there is a right angle turn in breeching, as well as into chimney, and the above coefficient becomes $\frac{13.6 + 9.4}{12}$, or 1.92 approximately.

89 For Case III, considering one right angle turn in breeching, the coefficient of d becomes 2.33.

90 In the foregoing it is assumed, as stated above, that the total flue area is 20 per cent greater than chimney area.

91 A study of the term $(H + 1.5\ d)$, equation [12], shows that for the conditions assumed, the resistance of flue and turn into chimney, both dependent on chimney diameter, is of as much influence on chimney performance as the resistance of chimney itself, whenever the height of chimney above the point of draft measurement is less than 12 by 1.5 or 18 diameters. Thus, in a 5-ft. chimney the above-named friction losses equal the chimney friction when chimney height above point of draft measurement is 90 ft. or less.

CHIMNEY PROPORTIONS FOR MINIMUM FIRST COST OF CHIMNEY

92 It is obvious that an indefinite increase in the diameter of chimney will not decrease the height below that necessary to maintain the draft corresponding to the fixed boiler resistances. An increase in height will reduce the diameter only to the point where the added draft per foot of height equals the chimney resistance per foot of height. Between these two extremes there are numberless combinations of height and diameter which will maintain the desired draft at base of chimney or the rear of boiler. But for a given set of conditions and for the type of construction under consideration there is but one height and one diameter resulting in a chimney of least first cost.

93 The simplest case is that for steel chimneys having a given average thickness of material, the cost of chimney being proportional to the product of height and chimney diameter.

94 For Case I, the final equation for minimum first cost, derived by the method of differential calculus, is reducible to simplest form. A chimney of least first cost, for the simple case stated, for a constant maintained draft inside chimney at chimney base, is that in which the frictional resistance of chimney is one-sixth of the potential draft of chimney.

95 This relationship was first stated in the book " Stirling," but incorrectly, as regards central chimneys as well as direct mounted chimneys, Case I. For Cases II and III this relationship will not apply, and no simple formula can be derived for these cases, owing to the resistance of breeching and chimney inlet, the latter involving terms of the fourth power. Resistance of chimney and breeching introduces terms of the fifth power, the final equation of minimum cost being too complex for ordinary engineering purposes. Equating the chimney resistance and one-sixth of potential draft of chimney we have

$$\frac{1}{6} fKH = \frac{2.3\ W^2 TH}{d^5} \quad \ldots\ldots\ldots\ldots [13]$$

It is apparent that the term H disappears, simplifying [13] to

$$\text{B.h.p.} = \frac{60^2}{G} \sqrt{d^5} \sqrt{\frac{\frac{1}{6} fK}{2.3\ T}} \quad \ldots\ldots\ldots\ldots [14]$$

96 The draft at base of chimney becomes, by the above relationship, five-sixths of potential draft of chimney. Column C, Table 1, gives chimney drafts on the above basis. Table 3 gives

TABLE 4—PART 1 APPROXIMATE MAXIMUM CAPACITY; ACTUAL BOILER HORSE-POWER; STEEL CHIMNEYS WITH SHORT FLUES; CHIMNEYS CENTRALLY LOCATED; STATIONARY B. & W. BOILERS AT RATING

BASED ON ATMOSPHERIC TEMPERATURE OF 80 DEG. FAHR.; SEA LEVEL, AND OTHER ASSUMED DATA. OIL FUEL

Ratio of actual boiler h.p. to rated boiler h.p. per cent 100
Draft in inches of water at boiler outlet, chimney side of damper 0.15
Corresponding excess air through boiler, per cent 50
Assumed excess air supply for determining boiler efficiency, chimney diameter and draft resistance of chimney and breeching, per cent 50
Assumed temperature of gases leaving boiler, deg. fahr. 450
Assumed temperature of gases entering chimney, deg. fahr. 425
Assumed boiler efficiency, working, not test conditions, per cent 75.4
Assumed lb. of chimney gases per actual boiler h.p. 52.8

Diameter, In.	Area, Sq. Ft.	Height in Feet above Boiler Room Floor Line											
		50	60	70	80	90	100	110	120	130	140	150	160
18	1.77	57	77	88	97	103	108	112	115	118	120	122	124
21	2.41	81	109	127	139	149	156	162	167	171	175	178	180
24	3.14	110	149	173	190	204	214	222	229	235	240	245	249
27	3.98	144	194	226	250	268	282	293	303	311	318	325	330
30	4.91	182	247	288	318	342	361	376	388	400	409	417	424
33	5.94	224	306	358	396	427	450	470	486	500	513	523	533
36	7.07	272	372	436	483	521	550	575	596	614	629	642	655
39	8.30	325	444	521	582	626	662	692	718	740	758	776	790
42	9.62	382	522	615	686	741	785	821	852	880	902	923	940
45	11.05	444	610	719	801	869	920	961	1000	1030	1060	1080	1110
48	12.57	512	702	830	927	1000	1060	1120	1160	1200	1230	1260	1290
54	15.90	660	911	1080	1210	1310	1390	1460	1520	1570	1610	1660	1700
60	19.64	827	1150	1360	1520	1660	1760	1860	1940	2000	2060	2110	2160
66	23.76	1020	1410	1670	1880	2050	2180	2300	2400	2480	2560	2630	2690
72	28.27	1220	1700	2020	2280	2490	2650	2790	2910	3030	3130	3210	3290
78	33.18	1450	2020	2400	2710	2970	3170	3340	3490	3620	3750	3850	3950
84	38.49	1690	2370	2830	3190	3490	3730	3940	4130	4280	4430	4560	4670
90	44.18	1960	2740	3280	3710	4060	4350	4600	4810	5000	5170	5320	5460
96	50.27	2250	3150	3770	4260	4670	5000	5300	5560	5780	5980	6160	6320
102	56.75	2560	3580	4290	4850	5320	5710	6050	6340	6610	6850	7060	7250
108	63.62	2880	4030	4840	5480	6030	6470	6850	7180	7500	7770	8010	8230
114	70.88	3220	4510	5440	6170	6780	7280	7720	8100	8440	8750	9050	9300
120	78.54	3580	5030	6060	6870	7560	8130	8630	9060	9440	9800	10100	10400
126	86.59	3960	5580	6710	7630	8380	9030	9590	10100	10510	10900	11300	11600
132	95.03	4390	6150	7420	8420	9260	9990	10600	11200	11700	12100	12500	12900
138	103.9	4800	6730	8150	9270	10200	11000	11700	12300	12800	13300	13800	14200
144	113.1	5230	7370	8920	10200	11200	12000	12800	13500	14100	14600	15100	15600
156	132.7	6210	8730	10600	12000	13200	14300	15200	16000	16700	17400	18000	18600
168	153.9	7190	10200	12300	14000	15500	16800	17800	18800	19700	20500	21200	21900
180	176.7	8350	11800	14300	16200	17900	19400	20700	21800	22900	23800	24700	25400

Effective height of chimney or height above point of draft measurement assumed 12 ft. less than tabulated heights corresponding to 12 high B. & W. boilers, 7 ft. furnace height. Draft resistance of boiler correct for 12 high boilers; capacities somewhat less for 14 high and greater for 9 high. Capacity reduced by long breeching. For correction for altitude, Table 5; for draft calculations, Table 2; for boiler efficiencies, Table 1; for capacity independent chimneys Table 3. When building conditions permit, select chimneys of least height in table for minimum

TABLE 4—PART 2 APPROXIMATE MAXIMUM CAPACITY; ACTUAL BOILER HORSE-POWER; STEEL CHIMNEYS WITH SHORT FLUES; CHIMNEYS CENTRALLY LOCATED; STATIONARY B. & W. BOILERS AT 50% OVERLOAD

BASED ON ATMOSPHERIC TEMPERATURE OF 80 DEG. FAHR.; SEA LEVEL, AND OTHER ASSUMED DATA. OIL FUEL

Ratio of actual boiler h.p. to rated boiler h.p., per cent 150
Draft in inches of water at boiler outlet, chimney side of damper 0.30
Corresponding excess air through boiler, per cent ..... Less than 50
Assumed excess air supply for determining boiler efficiency, chimney diameter and draft resistance of chimney and breeching, per cent 50
Assumed temperature of gases leaving boiler, deg. fahr. 525
Assumed temperature of gases entering chimney, deg. fahr. 500
Assumed boiler efficiency, working, not test conditions, per cent 73.0
Assumed lb. of chimney gases per actual boiler h.p. 54.6

Diameter, In.	Area, Sq. Ft.	Height in Feet above Boiler Room Floor Line								
		80	90	100	110	120	130	140	150	160
18	1.77	63	75	84	91	96	101	104	108	110
21	2.41	90	108	121	131	139	146	151	156	160
24	3.14	123	148	166	180	191	201	208	215	221
27	3.98	161	195	219	238	253	265	276	285	293
30	4.91	206	249	280	304	324	340	354	366	377
33	5.94	256	310	349	381	405	426	444	459	472
36	7.07	312	379	427	466	497	523	545	564	581
39	8.30	376	455	514	561	599	631	657	681	701
42	9.62	443	539	609	665	711	749	782	810	835
45	11.05	518	630	713	779	834	879	918	952	981
48	12.57	599	729	827	904	967	1020	1070	1110	1140
54	15.90	779	951	1080	1180	1270	1340	1400	1460	1500
60	19.64	985	1200	1370	1500	1610	1710	1790	1860	1920
66	23.76	1220	1490	1700	1860	2000	2120	2220	2310	2390
72	28.27	1470	1810	2060	2260	2430	2580	2710	2820	2910
78	33.18	1750	2150	2460	2710	2910	3090	3250	3380	3500
84	38.49	2060	2530	2900	3190	3440	3650	3840	4000	4150
90	44.18	2390	2950	3370	3720	4010	4260	4480	4670	4850
96	50.27	2750	3390	3880	4290	4630	4920	5180	5400	5610
102	56.75	3140	3870	4440	4900	5290	5630	5930	6190	6430
108	63.62	3550	4380	5020	5550	6000	6390	6730	7030	7300
114	70.88	3990	4920	5650	6250	6760	7200	7590	7930	8250
120	78.54	4440	5490	6310	6990	7560	8060	8490	8890	9240
126	86.59	4930	6100	7020	7770	8410	8970	9460	9900	10300
132	95.03	5450	6740	7760	8600	9310	9930	10500	11000	11400
138	103.9	5990	7420	8530	9460	10300	10900	11600	12000	12600
144	113.1	6550	8120	9350	10400	11200	12000	12700	13300	13800
156	132.7	7760	9630	11100	12300	13400	14300	15100	15800	16500
168	153.9	9060	11300	13000	14400	15700	16800	17700	18600	19400
180	176.7	10500	13000	15100	16700	18200	19500	20600	21600	22600

cost of chimney. Maximum chimney capacities stated are for uniform continuous load equally divided all boilers. For large plants subject to hand regulation, or for swinging load, reduce capacity from 10 to 20 per cent. If boilers have superheaters increase b.h.p. to include heat absorbed by superheater. Breeching 20 per cent excess stack area, without turns, length not exceeding 10 chimney diameters; boilers uniformly distributed along breeching; breeching depth = 1.5 × width. For capacities of chimneys with economizers, see Table 7. For economizer draft resistance, see Table 6.

capacities of economical chimneys for various assumed conditions, also heights for various required drafts. It will be noted that diameter is dependent only on boiler horsepower, and height dependent only on draft resistance of the boiler in question, information which must be supplied by the designer. Table 3 indicates chimneys of relatively large diameter but moderate height. Whenever conditions compel greater chimney height than given in table, the diameter may be reduced to maintain the required draft at chimney base, in accordance with equation [7].

97 The low chimneys indicated by Table 3 are noteworthy. For example: Assuming a Stirling boiler, requiring 0.1 in. draft at rating, with 450 deg. temperature, draft measurement 15 ft. above floor line, for 60 deg. atmosphere the height of chimney by interpolation is about 20 ft., to which must be added the 15 ft., making a total height of 35 ft. above floor. Under test conditions the draft in certain sizes is as low as 0.05 in., for which the height would be 11 ft. plus 15 ft. or 26 ft. above floor. At 100 per cent overload this total height would be increased to from 50 to 100 ft., depending on assumed draft resistance of boiler, atmospheric temperature, etc.

98 In general, it may be said that for Cases II and III the chimney of least first cost will be of moderate height, although this height will of necessity be somewhat greater than indicated by Table 3 for Case I. The least height in Table 4 corresponds to minimum first cost for the range of height intervals chosen. A cut-and-try method must be resorted to for greater refinement.

REDUCTION OF STACK CAPACITY AT ALTITUDES

99 The writer is not aware of any treatise dealing correctly with stack capacities at altitudes, either for coal or for oil fuel. Heretofore it has been assumed that the stack height increases inversely as the barometric pressure and that the stack diameter increases inversely as the two-fifths power of the barometric pressure, all other boiler and chimney data remaining constant. This relationship is based on the assumption of constant draft, measured in inches of water at the base of chimney for a given rate of firing of boilers, either expressed in lb. of coal per sq. ft. of grate surface or in ratio of rated capacity when burning fuel oil, regardless of altitude, an assumption wholly incorrect.

100 Whenever a chimney is working at maximum capacity

either at sea level or at an altitude, it is evident that the entire potential draft is utilized in overcoming the various resistances, each proportional to the square of the velocity, which varies at different points in the path of the gases.

101 Owing to the fixed areas along the path, all velocities may be related to a common velocity, say that within the chimney, and therefore all resistances in a given boiler plant for a given temperature of gases, etc., may be expressed as proportional to the square of the chimney velocity.

102 The total resistance to flow in terms of velocity head may be expressed in terms of the weight of a column of external air, and the numerical value of this head will be independent of the barometric pressure. Likewise the potential draft of a chimney, expressed in height of column of external air, will be numerically independent of the barometric pressure.

103 It is therefore evident that if a given boiler plant with connected chimney, operated with a fixed fuel, be transplanted from sea level to an altitude and if the temperatures remain constant (an assumption not entirely correct), the total draft head, measured in height of column of external air, will be numerically constant. The velocity of chimney gases will therefore remain the same at altitude as at sea level, and the weight of gases flowing per second with fixed velocity will be proportional to the atmospheric density, or inversely proportional to the normal barometric pressure.

104 Hence, except for errors due to assumption of constant temperatures of gases, it may be stated that the maximum capacity of a given boiler plant with a fixed size of chimney and fixed fuel will decrease at altitudes higher than sea level at a rate directly proportional to barometric height. For example: At an altitude of approximately 5764 ft., the normal barometric pressure would be 80 per cent of the barometer at sea level, and a chimney connected to a given plant, giving a maximum capacity of 1000 b.h.p. at sea level, would give not more than 800 b.h.p. at this altitude. However, corresponding to the reduction in rate of load on boilers, there would be a decrease in the temperature of boiler and chimney gases, tending under normal conditions to a still further reduction in maximum capacity.

105 An error commonly made in the consideration of stack capacities at altitudes is to assume that a given grade of fuel at a fixed boiler rating will require at an altitude the same draft,

measured in inches of water, at boiler damper as at sea level. It is evident that to develop a given boiler horsepower requires a constant weight of chimney gases and air for combustion. Hence, as the altitude is increased the density is decreased and, correspondingly, the velocity through the furnace, the bed of coal, or the firebrick checkerwork and the boiler passes, must be greater at altitude than at sea level. The mean velocity, therefore, for a given boiler horsepower and constant weight of gases will be inversely proportional to the barometric pressure, and the velocity head, measured in column of external air, will be inversely proportional to the square of the barometric pressure.

106 It is possible and usually desirable in coal burning to select a differently arranged boiler for altitudes, particularly in the West, increasing the relative grate surface not only because of reduction of draft loss, but also by reason of the poorer grades of coal, excessive ash content, etc. These considerations, however, do not apply with oil fuel, although it is possible but not desirable to modify the arrangement of the boiler, reducing somewhat its draft resistance.

107 A given chimney with gases at a fixed velocity, temperature, etc., will result in a draft head, measured in height of column of external air, numerically independent of the barometer. The absolute draft pressure, measured in inches of water, will therefore be inversely proportional to barometric pressure.

108 It follows that for chimneys operating at altitude, it is necessary not only to increase the height, but also the chimney diameter. Corresponding to the increase in height, there is an added resistance within the chimney, owing to the friction of the additional height. This frictional loss can be compensated by a suitable increase in the diameter, and when so compensated it is evident that the chimney height must be increased at a ratio inversely proportional to the square of the normal barometric pressure.

109 For Case I in which the chimney is mounted directly on top of the boiler, the gases entering the chimney without right angle turn, a simple calculation, based on equation [8], will indicate that the diameter of the chimney must be increased inversely as the two-fifths power of the barometric column. For example, for a height of 5764 ft. the normal barometer would be 80 per cent of that at sea level, and the height increases in the ratio of

1.5625 instead of 1.25 as is commonly assumed; the ratio of diameters will be 1.09.

110 If a chimney having height and diameter for minimum first cost be selected for sea level corresponding to a certain rating of boilers, the modification indicated for Case I will still result in the most economical chimney for operation at altitude.

111 For Cases II and III the effective chimney height may still be increased in inverse proportion to the square of the normal barometer, and the diameter of chimney and breeching must also be increased in such ratio that the modified chimney and breeching will have the same frictional resistance at sea level as before increasing the chimney height at sea level. In other words, the coefficient of resistance of chimney and breeching must remain a constant. In the general Case III, it was necessary to assume a relationship between flue length and chimney diameter; however, with the same boiler plant transplanted to an altitude, there is not, of necessity, any change in flue length. Therefore, if length and diameter of flue remain unchanged at altitude, its coefficient of resistance remains constant.

112 If the chimney diameter be increased at altitude in the ratio set forth for Case I and the breeching be increased to 20 per cent excess area, as for sea level, then with constant length of breeching the coefficient of resistance of both breeching and chimney inlet actually decreases, indicating that by applying to Cases II and III the same correcting factor as for Case I, the modified chimney will not only be ample, but slightly larger than necessary.

113 The general formulæ for altitudes are derived as follows: Let

P_o = normal barometric pressure at sea level
P_a = normal barometric pressure at altitude, and

$$R = \frac{P_o}{P_a}$$

Then if K, D_1, D_2, etc., still indicate values for sea level, and H and d new values at altitude, equation [7] becomes

$$\frac{fKH}{R} = R\left(D_1 + D_2 + \frac{2.3\ W^2TH}{d^5}\right) \quad \ldots\ldots\ldots\ldots [15]$$

also equation [8] becomes

$$\text{B.h.p.} = \frac{60^2}{G}\sqrt{d^5}\,\frac{\sqrt{\frac{fKH}{R^2}-(D_1+D_2)}}{\sqrt{2.3\,HT}} \quad \ldots\ldots\ldots\ldots [16]$$

Likewise equation [12] becomes

$$\text{B.h.p.} = \frac{60^2}{G}\sqrt{d^5}\,\frac{\sqrt{\frac{fKH}{R^2}-(D_1+D_2)}}{\sqrt{2.3T\,(H+1.5d)}} \quad \ldots\ldots\ldots\ldots [17]$$

114 Equation [17] assumes breeching length 10 chimney diameters, which is not necessarily the case, as pointed out above.

115 It is apparent that T is not modified by altitude, with the same rate of load on boilers and the same excess air.

116 Table 5 gives correcting factors for various altitudes.

MODIFICATION OF CHIMNEY CAPACITY DUE TO FUEL ECONOMIZERS

117 Owing to the comparatively small draft required when burning crude oil, it is unnecessary to resort to mechanical draft when installing fuel economizers.

118 It is self-evident that if economizers be added to an existing boiler plant, the available draft at boiler outlet will be decreased, not only by reason of the added frictional resistance of fuel economizers, but also because of the lowering of the temperature of gases within the chimney. It is also clear that the chimney dimensions must be modified to counteract these influences.

119 For the purpose of this paper, the writer has assumed the economizer dimensions given in the Green Fuel Economizer Company's catalogue. In determining the draft resistance of economizers there is involved not only the possible increased number of flue turns due to the installation of economizer, but also the frictional resistance of economizer tubes.

120 The writer has assumed that an economizer should be installed with an ample area for the flow of gases with the economizer "side dampers" closed, and the resistance of the economizers has been determined on this basis; the opening of side dampers then provides latitude for increased draft at boiler.

121 The resistance data for economizers indicate a lack of concordance, doubtless due to the fact that draft observations are usually made incidental to boiler and economizer tests and not for the specific purpose of noting the draft resistance of economizers. Owing to error of readings, the draft resistance of econo-

mizer being estimated as a difference of two draft observations, it is possible that the errors may be in opposite directions, tending to increase the actual draft resistance, or vice versa. Draft readings may be influenced by the velocity of flow of gases or by the relative heights of the two points of observation.

122 Gas analyses are usually taken at the boiler and not in the economizer setting, so that tests of record do not indicate closely the degree of excess air passing through economizer or the influence of air leakage in flues and economizer setting.

TABLE 5 CORRECTION FOR ALTITUDE; CHIMNEY CAPACITIES. OIL FUEL

Altitude, Feet above Sea Level	Normal Barometer	R Ratio Barometer Readings—Sea Level and Altitude	R^2 Ratio Increase Effective Stack Height	$R^{\frac{2}{5}}$ Ratio Increase Stack Diameter
0	30.00	1.000	1.000	1.000
1000	28.88	1.039	1.079	1.015
2000	27.80	1.079	1.164	1.030
3000	26.76	1.121	1.257	1.047
4000	25.76	1.165	1.356	1.063
5000	24.79	1.210	1.464	1.079
6000	23.87	1.257	1.580	1.096
7000	22.97	1.306	1.706	1.113
8000	22.11	1.357	1.841	1.130
9000	21.28	1.410	1.988	1.147
10000	20.49	1.464	2.144	1.165

123 Logically, the draft resistance of an economizer is proportional to the square of the velocity of flue gases, and to the length of economizer. The velocity of gases is directly proportional to the boiler horsepower, with an assumed excess air supply, and inversely proportional to the net area of the economizer, side dampers closed, and likewise proportional to the mean absolute temperature of gases within the economizer. The draft resistance of the economizer for sea level has thus been formulated as follows:

$$D_e = \left(\frac{\text{B.h.p.} \times T}{A}\right)^2 \times \frac{L}{T} \times \frac{J}{10^9} = \left(\frac{\text{B.h.p.}}{A}\right)^2 \frac{T \times L}{10^9} \times J \quad \ldots\ldots [18]$$

in which

D_e = draft resistance of economizer

B.h.p. = boiler horsepower actually developed, a quantity

proportional to the weight of gases flowing per minute

A = net area through economizer, side dampers closed

T = mean absolute temperature, deg. fahr., of gases within economizer

L = number of sections of tubes in economizer

J = a constant, assumed as 1.3

124 The quantity D is, in reality, a draft coefficient based on the fundamental quantities and, neglecting J, should be modified by a correcting factor, depending on the designer's data. Based on the writer's data, this coefficient averages about 1.3 and varies from 0.65 to 2.0; J is therefore made 1.3.

125 Table 6 gives approximate resistances for various economizers, based on boiler horsepower ratings corresponding to 100 and 150 per cent load on boilers and economizers, and indicates the assumed temperatures entering and leaving the economizer, ratio excess air, etc. It is important to note that the boiler horsepower indicated represents the actual boiler horsepower developed by the boiler and not the rated boiler horsepower, a quantity for purposes of estimating the resistance of the economizer strictly proportional to the actual weight of gases flowing per second.

126 The fundamental equation [6] of draft resistance is modified, due to installation of economizers, by the introduction of an added term covering the resistance of economizers and a correction of chimney temperature. The equation [12] for boiler horsepower for Case III, central chimneys, is then written

$$\text{B.h.p.} = \frac{60^2}{G}\sqrt{d^5}\sqrt{\frac{fKH-(D_1+D_2+D_e)}{2.3\,T\,(H+1.5d)}} \qquad [19]$$

where K is dependent on economizer outlet temperature.

127 Equation [19] is rendered approximate, as the term for breeching resistance is now made dependent on economizer outlet temperature; however, the breeching in a well designed economizer plant should be less than 10 diameters in length and these two errors may compensate. If added flue turns are introduced, [19] for close figuring must be corrected accordingly.

128 Owing to the special nature of economizer installations, variation in size and arrangement of economizers, etc., it is not practicable to compute a general table of chimney capacities for economizer plants. If a chimney of minimum first cost be selected

for the given boiler conditions, ignoring the installation of economizers, it should merely be necessary in providing for economizers to increase the height and not the diameter of chimney in order to compensate for the added resistance and lowering of stack temperature. By referring to the draft tables this increase in height may be determined approximately from the known increase in draft resistance and reduction of chimney temperature. It is not possible to compute any simple table setting forth accurately the ratios of increased chimney heights for the various stated conditions, based on chimney capacities without economizers, by reason of the involved breeching resistances, as well as stack resistances and varying ratios of total chimney draft required to overcome chimney friction.

129 It is a simple matter to compute for central chimneys a table of ratios giving the reduction of total boiler horsepower due to the installation of economizers, which, in connection with Table 4, enables a ready determination of chimney sizes. Table 7 has been computed on this basis and has reference to the capacities given in Table 4 for central chimneys without economizers.

130 It is self-evident that the minimum heights given in Table 4 will not furnish sufficient draft for any reasonable capacity, with the added friction and cooling of gases due to economizers. Table 7 indicates the minimum heights consistent with reasonable chimney proportions.

131 In order to obtain a chimney of minimum first cost when an economizer is used, trial and comparison should be made of several sizes.

GENERAL CONSIDERATIONS

132 The chimney capacities set forth in the tables are conservative for the exact conditions stipulated, although the margin for safety is not so great as might ordinarily be presumed from the approximate nature of chimney calculations.

133 These tables will not have application to all types of boilers and all conditions of operation and refer specifically to boilers of the highest type and under favorable conditions of operation. If the designer is uncertain as to the conformity of his conditions to those stipulated in tables, due margins for safety should be allowed for selecting chimney sizes.

134 While the writer has obtained results of operation for

TABLE 6—PART 1 APPROXIMATE DRAFT RESISTANCE, INCHES OF WATER; STANDARD GREEN FUEL ECONOMIZERS WITHOUT SIDE DAMPERS; 4 SQ. FT. OF ECONOMIZER HEATING SURFACE PER RATED B.H.P. OF BOILER, BOILERS AND ECONOMIZERS OPERATING AT RATING.

BASED ON 50 PER CENT EXCESS OF AIR SUPPLY; FEEDWATER TEMPERATURE 150 DEG., SEA LEVEL, AND OTHER ASSUMED DATA. OIL FUEL.

Draft Resistances in Inches of Water Column for various Widths and Lengths.

Temperature gases entering economizer, deg..... 425

Temperature gases leaving economizer, deg..... 275

Average temperature in economizer, deg..... 350

Gases per b.h.p. of boilers, lb..... 52.8

Assumed b.h.p. of economizer 3 b.h.p. per tube at rating.....b.h.p. $=3\ N=3\ (L\times W)\times 1.00$

Area	16.6		21.85		27.00		32.25		39.25	
Wide	4		6		8		10		12	
L	B.h.p.	*D'*	B.h.p.	*D'*	B.h.p.	*D'*	B.h.p.	*D'*	B.h.p.	*D'*
8	96	0.000282	144	0.000365	....	.....	....	.....	....	.....
12	144	0.000951	216	0.00123	288	0.00144	....	.....	....	.....
16	192	0.00225	288	0.00292	384	0.00341	480	0.00373	....	.....
20	240	0.00440	360	0.00571	480	0.00666	600	0.00729	720	0.00708
24	288	0.00761	432	0.00989	576	0.0115	720	0.0126	864	0.0122
28	336	0.0121	504	0.0157	672	0.0183	840	0.0200	1010	0.0194
32	384	0.0180	576	0.0234	768	0.0273	960	0.0299	1150	0.0290
36	432	0.0257	648	0.0333	864	0.0388	1080	0.0425	1300	0.0413
40	480	0.0352	720	0.0457	960	0.0533	1200	0.0583	1440	0.0567
44	528	0.0469	792	0.0610	1060	0.0709	1320	0.0776	1580	0.0754
48	576	0.0609	864	0.0792	1150	0.0920	1440	0.101	1730	0.0990
52	624	0.0774	936	0.100	1250	0.117	1560	0.128	1870	0.125
56	....	.....	1010	0.125	1340	0.146	1680	0.160	2020	0.156
60	....	.....	1080	0.155	1440	0.178	1800	0.197	2160	0.191
64	....	.....	....	.....	....	.....	1920	0.239	2300	0.232
68	....	.....	....	.....	....	.....	2040	0.287	2450	0.279
72	....	.....	....	.....	....	.....	2160	0.340	2590	0.331
76	....	.....	....	.....	....	.....	2280	0.400	....	.....
80	....	.....	....	.....	....	.....	2400	0.467	....	.....

DRAFT RESISTANCE

CORRECTION FACTORS FOR INCREASED AREA OF SIDE DAMPERS

No. of Dampers Open	No. of Tubes Wide				
	4	6	8	10	12
0	1.00	1.00	1.00	1.00	1.00
One Side	0.484	0.564	0.621	0.667	0.769
Both Sides	0.285	0.361	0.423	0.476	0.581

APPROXIMATE CORRECTION FACTORS FOR 75 PER CENT AND 100 PER CENT EXCESS AIR SUPPLY

Per Cent Excess.....	50	75	100
At Rating.....	1.00	1.34	1.74
At 1.5 Rating.....	1.00	1.34	1.74

TABLE 6—PART 2 APPROXIMATE DRAFT RESISTANCE, INCHES OF WATER; STANDARD GREEN FUEL ECONOMIZERS WITHOUT SIDE DAMPERS; 4 SQ. FT. OF ECONOMIZER HEATING SURFACE PER RATED B.H.P. OF BOILER, BOILERS AND ECONOMIZERS OPERATING AT 50 PER CENT OVERLOAD

BASED ON 50 PER CENT EXCESS OF AIR SUPPLY; FEEDWATER TEMPERATURE 150 DEG., SEA LEVEL, AND OTHER ASSUMED DATA. OIL FUEL.

Draft Resistances in Inches of Water Column for various Widths and Lengths.

Temperature gases entering economizer, deg. 500

Temperature gases leaving economizer, deg. 330

Average temperature in economizer, deg. 415

Gases per b.h.p. of boilers, lb. 54.6

Assumed b.h.p. of economizer 3 b.h.p. tube at rating ..... b.h.p. $= 3 N \times 1.50 \times 54.6 \div 52.8$.

16.6		21.85		27.00		32.25		39.25		Area
4		6		8		10		12		Wide
B.h.p.	D'	B.h.p.	D'	B.h.p.	D'	B.h.p.	D'	B.h.p.	D'	L
149	0.000732	223	0.000951	....	.....	....	.....	....	.....	8
223	0.00247	335	0.00321	447	0.00374	....	.....	....	.....	12
298	0.00586	447	0.00759	596	0.00886	745	0.00972	....	.....	16
372	0.0114	558	0.0149	745	0.0173	931	0.0189	1120	0.0184	20
447	0.0198	670	0.0257	893	0.0299	1120	0.0328	1340	0.0318	24
521	0.0314	782	0.0408	1040	0.0476	1300	0.0520	1560	0.0505	28
596	0.0468	893	0.0609	1190	0.0708	1490	0.0776	1790	0.0754	32
670	0.0667	1010	0.0868	1340	0.101	1680	0.111	2010	0.107	36
745	0.0915	1120	0.119	1490	0.138	1860	0.152	2230	0.147	40
819	0.122	1230	0.158	1640	0.184	2050	0.202	2460	0.196	44
893	0.158	1340	0.205	1790	0.239	2230	0.262	2680	0.255	48
969	0.201	1450	0.261	1940	0.304	2420	0.333	2903	0.322	52
....	.....	1560	0.326	2090	0.379	2610	0.416	3130	0.406	56
....	.....	1680	0.401	2230	0.465	2790	0.512	3350	0.497	60
....	.....	....	.....	....	.....	2980	0.619	3570	0.601	64
....	.....	....	.....	....	.....	3160	0.745	3800	0.724	68
....	.....	....	.....	....	.....	3350	0.884	4020	0.859	72
....	.....	....	.....	....	.....	3540	1.037	....	.....	76
....	.....	....	.....	....	.....	3720	1.212	....	.....	80

Table 6 has been computed by the use of the following empirical formula:

$$D = \frac{P_o}{P_1}\left(\frac{\text{B.h.p.}}{A}\right)^2 \frac{T \times L}{10^9} \times J$$

in which

D = the loss in pressure through the economizer expressed in inches of water column

P_o = normal barometer, sea level

P_1 = barometer at the altitude of plant

B.h.p. = B.h.p. rating of standard economizer as assumed above

A = net area in sq. ft. between tubes for the flow of gases

T = average absolute temperature in deg. fahr. of the gases in the economizer

L = the length of economizer in rows of tubes

J = a coefficient determined from tests

For the purposes of this table the value of J is taken as 1.3, an average of values from a number of tests ranging from about 50 per cent above to 50 per cent below this figure. For 75 per cent and 100 per cent excess air supply, see approximate correction factors. If the volume of excess air due to leakage is sufficient to change the temperatures assumed, the correction factor for excess air will be correspondingly changed. B.h.p. refers to work done by boilers only, based on feedwater temperature entering the boilers, and not that entering economizers.

Draft resistance is based on areas of standard economizers with standard spacing and tube length. No allowance has been made for the resistance of any additional breeching or turns made necessary by the use of the economizer. Such resistances must be added to the values given in table.

For altitude D should be increased inversely with the barometric reading.

See footnotes, Table 7, regarding air supply greater than 50 per cent excess, leakage through brick settings, etc.

TABLE 7 SHOWING APPROXIMATE RATIOS OF CAPACITY OF CHIMNEYS WITH ECONOMIZERS TO CAPACITY OF SAME CHIMNEY WITHOUT ECONOMIZER BASED ON ASSUMPTIONS BELOW

	At Rated Load									150 Per cent of Rated Load								
Initial temperature of gases in stack without economizer, deg.	425									500								
Initial temperature of gases in stack with economizer, deg.	275									330								
Db. Draft at chimney side of boiler damper, in.	0.15									0.30								
D_e	0″	0.005″	0.010″	0.025″	0.05″	0.10″	0.15″	0.20″	0.30″	0″	0.005″	0.010″	0.025″	0.05″	0.10″	0.15″	0.20″	0.30″
H	Ratio of Chimney Capacities									Ratio of Chimney Capacities								
60	0.502	0.439	0.365															
70	0.669	0.638	0.608	0.500	0.233													
80	0.737	0.717	0.696	0.630	0.500													
90	0.774	0.759	0.743	0.694	0.605	0.364				0.495	0.455	0.411	0.233					
100	0.798	0.786	0.774	0.735	0.667	0.498	0.233			0.645	0.623	0.600	0.526	0.371				
110	0.815	0.805	0.794	0.762	0.705	0.575	0.406			0.715	0.699	0.684	0.634	0.542	0.275			
120	0.827	0.819	0.810	0.782	0.734	0.628	0.497	0.321		0.759	0.748	0.736	0.698	0.630	0.465	0.189		
130	0.837	0.830	0.822	0.798	0.756	0.665	0.560	0.428		0.788	0.778	0.769	0.738	0.684	0.561	0.401	0.080	
140	0.844	0.838	0.830	0.809	0.772	0.693	0.603	0.498	0.126	0.809	0.801	0.795	0.767	0.721	0.623	0.504	0.347	
150	0.851	0.844	0.838	0.818	0.785	0.715	0.637	0.548	0.299	0.825	0.818	0.811	0.789	0.750	0.666	0.571	0.456	
160	0.855	0.850	0.844	0.826	0.797	0.733	0.664	0.586	0.388	0.837	0.831	0.825	0.805	0.771	0.699	0.619	0.526	0.254

The capacity ratios given above are applicable to chimney capacities given in Table 4 for boiler plants without economizers, chimneys centrally located sea level, and are subject to all of the limitations stated for Table 4. D_e = draft resistance due to economizer, for values of which see Table 6. If economizer layout introduces greater breeching resistance than assumed in development of Table 4, D_e should be increased accordingly to cover additional length of and turns in breeching. Table 4 is based on straight run of breeching. B.h.p. refers to work done by boilers only, based on feedwater, temperature entering boilers, and not that entering economizers. Boiler efficiencies are as stated in Table 4, not including economizer saving. Ratios given for various heights enable the maintenance of same net draft at boiler outlet for stated boiler ratings, making necessary allowance for loss of draft due to cooling of chimney gases and interposed economizer resistance. For sake of uniformity with other tables, ratios are based on 50 per cent excess air supply at maximum load measured in chimney. This condition is realizable in small plants having masonry settings and in large plants with steel breechings and steel economizer casings. For large plants, with brick flues and brick economizer settings, air leakage may increase relative volume of chimney gases from 20–40 per cent or more. D_e should be modified accordingly for resistance, and the then calculated b.h.p. of chimney reduced to allow or air leakage. Correction must also be made for modified chimney temperature.

numerous chimneys burning oil fuel, it is generally the rule that oil-burning chimneys in actual service are underloaded and that comparatively few chimneys are operating at their maximum capacity. Therefore, in computing chimney capacities given, it has been the writer's aim to determine from test observations the actual boiler resistances for the various conditions of operation, and from this starting point to calculate chimney height sufficient to produce the required draft at boiler and diameter sufficient to carry away the necessary volume of gases, with the margin of draft available for the friction of flow through breeching and chimney.

135 Where building conditions demand chimneys of considerable height, the draft required at the boiler outlet becomes a smaller proportion of the potential draft of the chimney, and accordingly an error in the assumption of the draft resistance of the boiler will have less effect on the total capacity of the chimney of given dimensions.

136 When selecting chimneys of reduced height for boilers operating at or near rating, the draft resistance of the boiler for least cost and economical operation requires a larger proportion of the potential draft of the chimney. An error in the assumption of draft resistance of the boiler largely affects the capacity possible with a chimney of given dimensions.

137 In all instances the diameter of the chimney and frictional loss within the chimney and breeching have been based on 50 per cent excess air over chemical requirements. For a number of years the writer thought it necessary to provide for not less than 100 per cent excess air; but from actual experience and data from a number of tests in which the firing conditions were directed by ordinary operators, it has been shown that 50 per cent excess air is sufficient for good practice under maximum load conditions in first class installations.

138 The various features affecting the draft resistance of a boiler have been pointed out, many of these being subject to variation by the local operator. The greatest influence on draft resistance, however, is that due to excess air. Under ideal conditions it is possible to fire boilers with approximately 10 per cent excess air and even less, giving a corresponding reduction in the draft resistance of the boiler. As the excess of air supply is reduced, the capacity of a chimney increases in a multiplied ratio, due not only to the decreased volume of gases to be handled and

the decreased weight of gases per b.h.p. by reason of greater boiler efficiency, but also to decreased draft resistance of the boiler and the greater draft available for the flow of gases through breeching and chimney. It will therefore be possible not only to point to chimneys of the dimensions given in tables which show a greater capacity than stated corresponding to reduced air supply, but also of less capacity than stated corresponding to increased air supply.

139 In considering boilers of differing types, the chimney dimensions will depend largely on the boiler design and efficiency, particularly with respect to the draft resistance of the boiler and the minimum air supply as dependent on furnace design. Certain types of boilers will show less draft resistance and others more than indicated for Babcock and Wilcox boilers, with a corresponding variation in chimney capacity.

140 Many persons inexperienced in the burning of crude oil have the impression that the mere closing of the damper will result in the reduction of excess air supply practically to the theoretical limit. It is actually possible to cause the formation of smoke with considerably more than 50 per cent excess air supply in a poorly designed boiler furnace. The furnace must have a particular shape, volume, and arrangement of checkerwork with respect to the burner to permit the complete admixture of air for combustion and vaporized oil; and the furnace efficiency with respect to this admixture, influenced by the surrounding heated brickwork, determines the minimum excess air permissible without undue smoking.

141 In the calculation of chimney tables it was deemed best not to provide sufficient margin for safety in order to cover all possible conditions of operation, but rather to establish as a benchmark the maximum chimney capacities based upon the draft resistances obtained during conditions of uniform load, and to modify capacities as need be for the special conditions of operation, which should be carefully studied before selecting chimneys for a given installation.

142 When operating with 50 per cent excess air, uniform load, and furnaces properly adjusted, there should be a slight haze from each boiler; on uniform load there should not be an objectionable amount of smoke from central chimneys connected to a number of boilers, except when fires are unequally adjusted.

As is well known, automatic control of fires assures an equal division of load among all boilers in service.

143 It is possible for short intervals to force the chimneys beyond their maximum capacity, giving rise to increased smoke density; but good practice does not sanction such heavy smoking, except for the inevitable short intervals on swinging loads.

144 In passing, it may be stated that it is common practice to gage the air supply roughly by the density of smoke emitted from the chimney, and it is wise in boiler plant design to provide openings through the roof so that the firemen can conveniently observe the chimneys.

145 The tables provide for the slight reserve necessary to regulate fluctuating steam pressure with careful firing, but do not provide sufficient reserve to restore steam pressure rapidly after a material pressure drop. During such intervals careless firing may require a very large momentary overload on boilers. While a certain draft margin is necessary, a chimney which makes necessary careful attention of furnaces is a positive benefit from the standpoint of boiler abuse and cost of tube renewals.

146 If a number of boilers be connected to a central chimney subject to hand-firing, certain boilers under unequal adjustment of fires will be operating at overload and others at underload. To offset these conditions and to maintain the maximum total capacity required, it is necessary slightly to increase the height of stack, and in the table no margin is specially provided to cover this contingency. When firing under automatic control, which insures an equal division of load among all boilers in service, such margin for increase is unnecessary.

147 It has been found in large plants having central chimneys and subject to heavy swinging loads, that on a sudden increase of load there is a sluggishness of chimney action due to inertia of gases within the chimney and breeching, as well as a time element required for the heating of the chimney; smoke is usually formed during such intervals. Particularly where flues are of masonry construction is it desirable to increase the height of chimney slightly in order to counteract this influence, although occasional smoking cannot even then be completely eliminated.

148 For swinging loads the writer would reduce the tabulated capacities for central chimneys from 10 per cent to 20 per cent, this factor being one of judgment and not based on experimental data.

149 When boilers have been operating for a period at reduced load, and a sudden increase in load takes place owing to the cooling action of the furnace, an increased air supply will be required over uniform load conditions in order to obtain the desired capacity without smoking. This condition will continue until the furnace brickwork has been thoroughly heated, and common practice permits smoking during this short interval.

150 Consideration should be given to a certain reserved draft in order to insure, at all loads, a positive draft within the setting by throttling the air admission. It has been pointed out by Dr. Jacobus that under certain load conditions, should a negative draft within the setting occur, there will be a tendency for the hot gases to pass outward through the setting, making possible a considerable overheating of the brick furnace. On the contrary, with a positive draft within the setting, a certain amount of air infiltration will tend to cool the brickwork and thus increase the life of the setting. In general, the tendency to form a negative draft diminishes with overload.

151 In the design of chimneys it is necessary to give careful consideration to the question of minimum air supply by reason of the tendency of increased smoking with changes of load when operating near the theoretical limit. Sooting of boiler tubes might easily reduce the heat transfer of the boiler to such an extent as more than to offset the economy due to decreased air for combustion.

152 A well designed central chimney, subject to variation of number of boilers in service, requires, for maximum economy, a comparatively small amount of stack and breeching friction in order to insure what might be termed "draft regulation." In boilers subject to hand adjustment or, for that matter, automatic control of air supply, it is desirable to have a nearly fixed draft at the boiler outlet for each addition of load, with the least possible change due to variation in the number of boilers in service. It is self-evident that this variation in draft will give a variable air supply, with a fixed position of damper for each load, and thus reduce the fuel economy of the plant.

153 The proper determination of chimney height, with respect to the maximum draft required at boilers, will have a marked influence on the fuel economy of the plant. While a certain draft margin is necessary for the special conditions of plant operation, it should at all times be the aim to provide a chimney

with the least possible excess draft over the necessary maximum.

154 In plants subject to uniform factory or pumping loads approximately at rating, the chimney height should be very much reduced and such that, when boilers are operated with nearly wide open dampers, the air supply will not materially exceed the requirements for economical firing.

155 With excessive height of chimney it is not possible to open the damper more than a fractional amount, thus giving a large increased air supply for a very small added opening of damper, making economical operation at variable loads impossible with hand control, and more difficult with automatic control.

156 The influence of excessive draft becomes more apparent when boilers are operated for intervals at reduced capacity. If the firemen be careless and leave the dampers and ash doors unchanged during variations of load, it is apparent that the draft resistance of the boiler becomes nearly a constant, and except for a slight decrease in the temperature at reduced loads, the tendency is toward a practically constant flow of gases through the setting, regardless of the rate of firing, and, correspondingly, an increasing ratio of excess air at light loads. While with tall chimneys the excess air supply is abnormal even at maximum loads, it becomes thus increasingly wasteful on reduced loads.

157 In one instance which came to the writer's observation, a chimney gave 300 per cent excess air when operating at rating, and 450 per cent excess air when operating 30 per cent load, the dampers and ash doors being wide open in both instances. In the latter instance the resulting boiler efficiency was less than 40 per cent, whereas, even at this reduced load, the efficiency should have been approximately 75 per cent.

158 Experience has indicated that plants having tall chimneys subject to hand control are, in general, uneconomically operated; while plants having relatively low chimneys indicate a superior performance, due largely to the decreased margins for excess draft and possible wastage of fuel with excess air supply. This fact is particularly impressive after an examination of the economy attainable in power plants in connection with tall office buildings, where high chimneys are a necessity. In such instances a reduction of stack diameter to absorb by stack and breeching friction a considerable portion of the excess draft will result in an improved economy of plant.

CHIMNEYS FOR AUXILIARY STEAM POWER STATIONS

159 It has been pointed out that chimneys of reduced height are possible under certain conditions of operation, namely, with uniform load, approximately at rating, but the impression should not prevail that tall chimneys are unnecessary for oil firing under certain conditions of heavy load.

160 In auxiliary steam power stations it will be found necessary at times to force boilers to approximately 100 per cent overload. Under these conditions, due to the rapid increase of draft resistance of boiler, the heights of central chimneys will vary, depending on the type and size of boiler, plant arrangement, etc., from less than 100 ft. to about 150 ft. for sea-level conditions, and even greater heights will be required at altitudes.

161 The height may be somewhat reduced below that otherwise necessary on heavy overloads by increasing judiciously the admission area through floor of furnace. On large boilers, reduction of draft and chimney height at overloads may also be effected by the provision of larger air admission doors, limited, however, by the creation of a negative draft within setting, as already discussed.

162 It is important to note that the chimney heights stated for certain overload capacities apply only when a sufficient number of the connected boilers are in operation to maintain the stated initial chimney temperature, and to prevent undue cooling within the chimney. In auxiliary steam power plants, or other stations where at times comparatively few of the boilers may be in operation on a central chimney, this feature must be considered.

163 If the temperature of the water is, say, 100 deg. fahr., or less, within a boiler connected to a cold chimney, a considerable period will elapse on an emergency call before the gases entering the chimney will reach such temperature as to produce an appreciable draft and make possible rapid firing. If, as is ordinarily the case, all the boilers be kept under steam by intermittent and alternate firing, the temperature of the boiler heating surface will be considerably above the boiling point, and the mere opening of the boiler dampers will cause a flow of air through the setting, the incoming air being heated by the hot water within the boiler tubes. The starting of the fire will grad-

ually increase the temperature of the escaping gases, but owing to the reduced draft, a considerable interval of time will elapse before the boiler will be operating at such rate of firing as to give the necessary increase in temperature of escaping gases and chimney draft for any appreciable overload on the boiler. Under these conditions it is evident that the taller the chimney the less the required time for getting under way; also, that an independent chimney will, in general, enable quicker starting than a central chimney when firing a fractional number of boilers. On the other hand, if a central chimney be employed, and one or more boilers are constantly under fire, the chimney will at all times have a relatively higher minimum temperature, and by reason of the draft produced, enable the quicker starting up of a large number of connected boilers than is possible with independent chimneys.

164 As service demands become more exacting it is probable that auxiliary steam power stations will ultimately be provided with forced draft fans to supply air for the rapid starting of boilers which are to be used for the first few minutes, and until boilers, furnaces, flues and chimneys are thoroughly heated, thus following, to a certain extent, battleship practice; or steam jet chimney blowers may be used while starting.

EXAMPLES FROM PRACTICE

165 *Plant No. 1.* A plant of three Babcock and Wilcox boilers, 7 high, total 275 h.p.; one Green fuel economizer 6 by 20; one side damper; one pair of induced draft fans; one 42-in. discharge stack; 54-ft. height above floor line. Plant initially installed for coal burning. When fired with oil fuel, at rating and at slight overloads it was found that the natural draft from chimney was sufficient to produce the necessary draft at boilers to overcome resistance of economizers and resistance of blades of fan wheels with fans at rest. When a wind storm reduced the height to 40 ft. the capacity was still ample. A second wind storm reduced the height to 19 ft. which was ample at light loads, but required assistance of fans at 180 r.p.m. at rating.

166 *Plant No. 2.* A plant having a number of 9 high Babcock and Wilcox boilers; after the San Francisco earthquake of 1906, was able to operate at rating with chimney height reduced by earthquake to 25 ft. above floor line.

167 *Plant No. 3.* A plant of 12 high Babcock and Wilcox boilers, with horizontal short breeching and steel chimney 115 ft. above floor line; sea level; chimney diameter relatively large and friction small; three boilers connected to one chimney, one boiler, on test, operated at 210 per cent of rated capacity, test conditions.

168 *Plant No. 4.* A plant of 12 high Babcock and Wilcox boilers; a total of 15 boilers, connected to a central chimney 125 ft. above floor line; Green fuel economizers; induced draft fans; one boiler operated at 208 per cent of rating, other boilers at about rating. The ordinary draft of chimney was insufficient, and necessitated the use of fans. Economizers 80 tubes long, resistance excessive.

169 *Plant No. 5.* A plant of 12 high Babcock and Wilcox boilers, 8 boilers connected to a central chimney; Green fuel economizers; panel system. Chimney reduced by earthquake to 70 ft. Natural draft sufficient for operation of boilers at about rating, but insufficient for overload.

170 *Plant No. 6.* A plant of 14 high Babcock and Wilcox boilers; central chimney; large diameter; approximately 70 ft. height above floor line. Maximum capacity one boiler under test, when other boilers operating at about rating, natural draft 145 per cent of rating.

171 *Plant No. 7.* Stirling boilers, 250 h.p., connected to independent overhead chimney 80 ft. height above floor line. Maximum recorded capacity, but not maximum possible capacity, 90 per cent overload, still showing an excess air supply; plant operating factory load, boilers intended for operation at rating only; on account strong draft from chimney could not get fireman to operate boilers at rating. Finally reduced chimney height to 40 ft. to prevent such excess air through furnaces as would prevent excessive overloads on boilers. In this manner capacity considerably reduced, but still able to obtain greater capacity than rating. Exact overload not determined. Plant economy increased by lowering the temperature of escaping gases and amount of excess air.

172 *Plant No. 8.* Battery of two Stirling boilers 250 h.p. each, connected to a central overhead chimney with A-shaped breeching, the total height being 40 ft. above floor line. Intended for use in oil fields with unskilled firemen. Height fixed at 40 ft. to prevent overload on boilers and the waste of fuel from excessive overload and from excess air. Boilers have operated at capacities in excess of rating.

173 *Plant No. 9.* Office building plant. Stirling boilers, chimney height approximately 200 ft. On account of excess draft, even with partially open dampers and air doors excess air over 200 per cent at rating; probably 15 to 20 per cent waste of fuel due to excess air.

DISCUSSION

William Kent. In regard to oil burning, the writer has had an opportunity to make a test with a Scotch marine boiler having four furnaces, each with one oil burner. The test was under forced draft with the furnace and ashpit doors closed so that the flame could not be seen; one of the burners got out of order and smoked badly for a long time before its condition was discovered.

It appeared that when more than one burner was in use, one of them might be running with too much air and another with too little, both giving poor economy, while the average flue gas

analysis would indicate that the air supply was just right. The writer was looking for an apparatus resembling a steam gage, with a pointer to indicate the percentage of oxygen in the chimney gases, and asked if anyone knew of such. If the combustion chamber and flues beyond it were so divided that each section of the chamber were fed by one burner, the indicator would show whether or not each burner was working under ideal conditions. The condition of successful operation of either oil or coal-burning furnaces was, as far as he was able to find, that the oxygen in the gases should be between 3 and 5 per cent. hot air would uselessly be carried out of the chimney. The apparatus he wanted would tell him when he had between 3 and 5 per cent of oxygen in the gas.

E. H. Peabody referred to Mr. Kent's statement that it is possible to have two burners in operation, one with too much air and one with too little. The furnace under consideration is designed to take care of a poor mixture, and it is quite possible to take care of any irregularities of that kind: but of course it is a mistake for one burner to get too much air, as that can be regulated. The importance of the furnace in oil burning is much greater than that of the burner.

D. S. Jacobus. Mr. Kent's idea of determining the oxygen component in the flue gases in place of the carbon dioxide is a feature that is not as important as the accurate determination of the carbon monoxide. Mr. Kent says that 3 per cent of oxygen is a good amount in oil burning, but this is not all that must be striven for, as with three per cent of oxygen a poor furnace may allow considerable carbon monoxide to escape to the stack, where in a good furnace there will be no carbon monoxide, or at best not over one-fifth of one per cent.

The Author. Mr. Kent's discussion refers to the question of oil burning, rather than directly to chimney capacities, which is the subject matter of this paper. It appears that Mr. Kent has been fully answered by Dr. Jacobus and Mr. Peabody.

In view of the recommendations in this paper in favor of low chimneys for boilers operating at uniform load, it is advisable to repeat a well-known caution in regard to the effects of downward currents of the air during high winds when the chimney is unfavorably situated in respect to surrounding buildings, hills, etc.; with a low chimney the loss of draft from this cause is

more pronounced than with higher chimneys. Based on the formula for pressures due to wind velocity given in Kent, the writer has computed the corresponding pressure in inches of water for various velocities as follows:

Wind Velocity, Miles per Hour	Equivalent Pressure, Inches of Water	Condition of wind
1	0.0005	Hardly perceptible
2	0.002	Just perceptible
3	0.0044	
4	0.0078	Gentle pleasant wind
5	0.0122	
6	0.0176	
7	0.0176	
8	0.0239	Pleasant brisk gale
9	0.0395	
10	0.0489	
12	0.0705	
14	0.0955	
15	0.11	
16	0.125	
18	0.159	Very brisk
20	0.195	
25	0.306	
30	0.447	High wind
35	0.6	

These pressures are realized on a plain surface at right angles to the direction of the wind. The vertical component of wind velocity would be effective only in the case of boiler chimneys.

Since the preparation of this paper, data have been secured on a rather interesting oil burning plant, the results of which appear to be quite ideal. The plant in question is the Sloat Boulevard pumping station of the Spring Valley Water Works, San Francisco. Two Babcock & Wilcox boilers are installed, each rated 152.5 h.p., also a Green fuel economizer having a total of 16 sections, each containing six tubes 4 in. in diameter. The whole is connected to a chimney 66 in. in diameter, having a top height of 35 ft. above boiler room floor line.

The chimney has a relatively large diameter for the work and the draft loss by friction is therefore minimized. There is one right-angle turn from the boilers entering the economizer with a straight pass from the economizer to the chimney, except a right-angle turn in passing upward into the chimney.

The station records of water and oil measurement indicate that the boilers operate at about 10 per cent above rating, one boiler only being in service at a time.

In order to obtain sufficient air for combustion, it is necessary to keep the front ashpit doors, rear boiler damper and stack damper wide open. While no observations were made of the gas analysis, it appeared that combustion was proceeding with the usual minimum air supply and without smoke. If such is the case, it is apparent that when operating at this load, the most negligent fireman could not materially increase the air supply for combustion, because all air doors and dampers are wide open and there is no means of increasing the draft.

The author has suggested for trial purposes, the further reduction in stack height by the removal of one course of steel, but this has not as yet been tried. The writer knows of no better example than this for the regulation of boiler efficiency by stack height; this regulation, however, is limited to plants operating under practically uniform load conditions.

No. 1362

TESTS OF A 1000-H.P. 24 TUBES HIGH B. & W. BOILER

By B. N. Bump, Syracuse, N. Y.

Member of the Society

The boiler tested was erected as an experiment for the purpose of determining, by test and by continuous operation in the regular service of a large plant, the advantages and disadvantages of so high a boiler of the B. & W. type. These tests were made from November 1911 to February 1912. They were expected to answer the following questions:

a At what rating will the highest efficiency be obtained?

b Will the superheat obtained, with the superheater located above the 24th tube, be sufficient to pay for the installation of the superheater?

c Will the last pass of the boiler be effective?

d Can the exit gas temperature be reduced very nearly to the temperature of the steam in the boiler?

e If the exit gas temperatures are close to the steam temperature, will pitting occur in the back end of the boiler due to the sulphur content of the gas?

BOILER AND SUPERHEATER

2 The boiler was built up of one regular 14-tube section and above this, with a space of some 11 in. between them, was placed a 10-tube section. The headers of the 10-tube and 14-tube sections were joined by short nipples, making a 24 high header. There are 21 of these sections containing a total of 504 tubes. The tubes are 4 in. in diameter, and 18 ft. long. There are 3 drums 42 in. in diameter, and 24 ft. long. The water heating surface of the boiler is 10,000 sq. ft. Between the top tubes and

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the drums a Foster superheater is placed containing 1750 sq. ft. of superheating surface.

3 The gases make three passes through the boiler. The baffles are arranged to give a gradual decrease in space through the first and second passes in the direction of flow, which tends to maintain the velocity of the gases as the temperature decreases. The gases enter the superheater from the top of the first pass, after traveling over 24 tubes. From the superheater they turn downward through the second pass; upward through the third pass, and then make their exit between the downtakes, and through the rear wall to the stack. The spaces between the headers, both front and rear, were packed with asbestos.

STOKER AND FURNACE

4 The boiler is fired by a six-retort Taylor stoker, grate area 62.465 sq. ft., not including any dump grate area. The ratio of boiler heating surface to grate area is 160 to 1. The ratio of boiler plus superheater surface to grate area is 188 to 1.

5 The height of the combustion chamber is about 6.5 ft., and the width of the furnace 12 ft. 7 in., which latter is greater than the width of the stoker to allow for placing a cast-iron wind box on either side of the stoker. This is believed to be a new feature, tried as an experiment. Previously there had been trouble from the formation of clinker on the side walls with stokers of both the underfeed and overfeed type, owing to some coals giving a tough sticky clinker which is very troublesome. With the side walls against the stoker, clinker forms rapidly, and is difficult to remove. Setting back the side walls lessens clinker formation to some extent, and facilitates the removal of the clinker.

6 A brick ledge between the stoker and side wall is objectionable, because in time clinker will fasten to it and it is almost impossible to get a bar between clinker and ledge. The cast-iron wind boxes overcome the objections to the brick ledge by chilling the clinker so that it does not stick. A bar can always be forced between the clinker and wind box to raise the clinker. A part or all of the air from the blast fan is passed through these wind boxes on its way into the stoker. After six months of continuous service there is no sign of burning on the wind boxes.

7 Fig. 1 shows a front and side elevation of the unit. On the side elevation the locations of the pyrometer rods and gas sampling tubes are shown. The furnace temperatures were taken

through the three openings shown just under the lower tubes. The other gas temperatures were taken at positions 1 to 7 inclusive. The numbers on the data sheets and charts of gas temperatures refer to these positions.

8 The walls were lined with 9 in. of fire brick backed up by 9 in. of porous insulating brick. The outer surfaces of all walls were painted with two coats of boiler pitch to prevent any filtration of air through the brick work. Against these painted surfaces of the front and side walls an 85 per cent magnesia covering 4 in. thick was placed. The magnesia was held against the brick and protected by a thin steel shell. Asbestos board was placed on the top half of the drums, then loose mineral wool was placed over the entire top of the boiler.

TEST APPARATUS AND DATA

9 The tests varied from 18 to 48 hours in length, although most of them were about 24 hours long. The coal was weighed in a wheelbarrow on scales tested frequently for accuracy. A small sample of coal was taken from each wheelbarrow load to make up the test sample for moisture, analysis, and calorific value.

10 The feedwater was measured in a tank of 13,000 gal. capacity. All feedwater piping was under observation, and any leaks were quickly discovered. The drips from the pump plungers were caught and returned to the feed tank. The feedwater piping being entirely independent there was no possibility of the measured water going anywhere except into the boiler. The blow-off piping was slip blanked, and the boiler blown down only between tests. The boiler was examined for leaks at frequent intervals, and the boiler tubes were dusted every 48 hours in the intervals between tests.

11 Steam pressures, superheated steam temperatures, and steam calorimeter temperatures were taken by recording instruments.

12 It was found necessary to have some suction over the fire to prevent the escape of gas into the boiler room. This suction was kept as low as possible. Gas temperatures were taken at seven different points as given in Table 4. The seven pyrometer rods were wired to one indicating instrument so that, by the use of knife edge switches, the indication of one rod after another could be taken in quick succession.

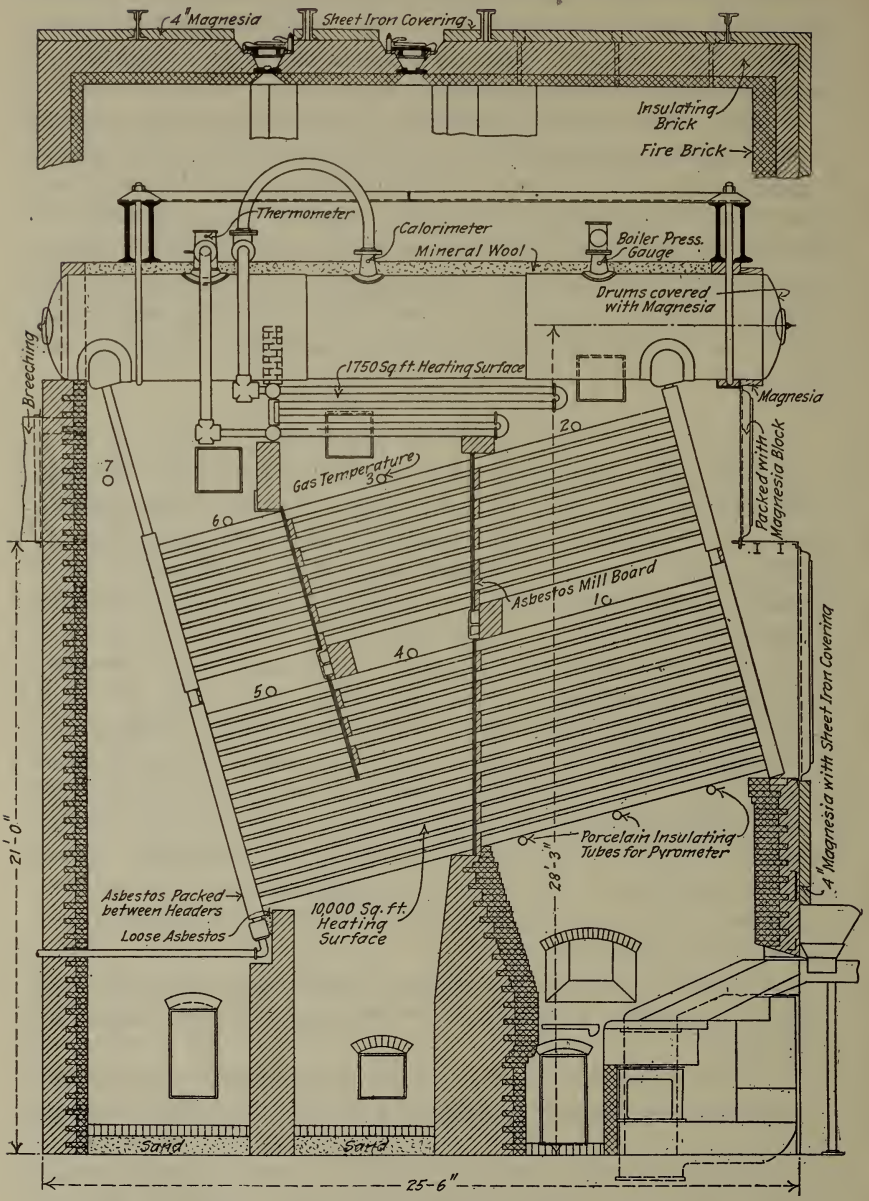

FIG. 1-a LONGITUDINAL SECTION OF BOILER

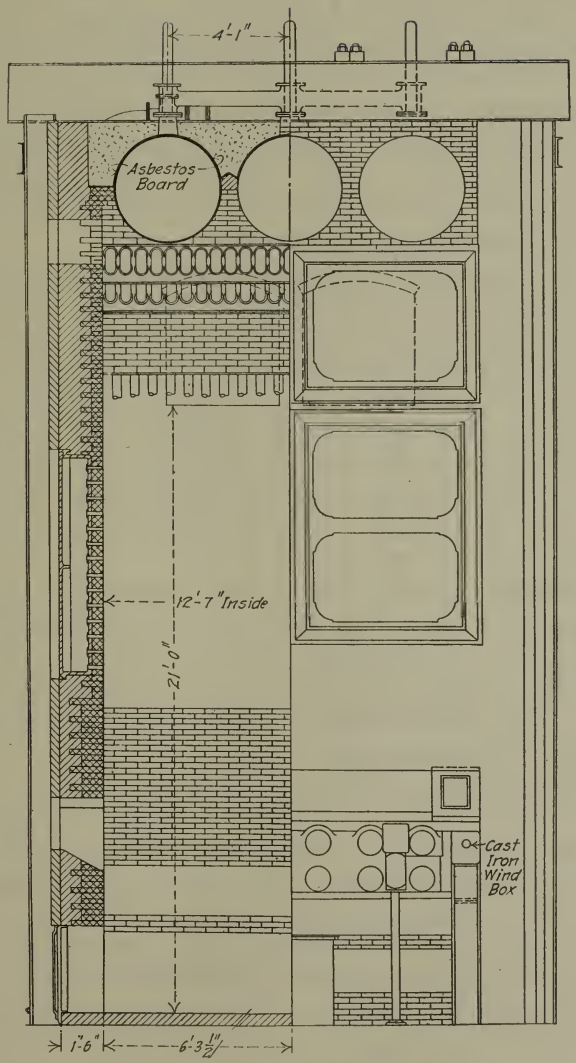

FIG. 1-b FRONT ELEVATION OF BOILER

13 A high reading pyrometer was used for furnace temperatures. There was considerable trouble in keeping this pyrometer in working order and consequently the furnace temperatures are given for only a few of the tests.

14 Samples of gas for analysis were taken in the first pass at the top of the 14th tube, and before the breeching. The samples were taken through six sampling tubes placed in the space between the boiler and back wall, and directly in front of the opening to the breeching. One tube was connected to a CO_2 recorder and the other five were attached to large sampling bottles which were run continuously. The tips of the tubes were so placed that the samples taken through each represented approximately equal volumes passing to the stack.

15 The first 19 tests were made with mixed coal, from four different mines all in the same region. Tests Nos. 21 to 40 were all run on coal from No. 29 mine, lighter than that used on the early tests and giving much less clinker trouble. It made a loose fuel bed and allowed more boiler capacity with less draft in the stoker. For best combined efficiency the dry coal per square foot of grate per hour is about 26 lb., or nearly 270 lb. per retort. The quantity of coal burned per square foot of grate per hour was increased to nearly 63 lb. with a decrease in combined efficiency of 5.5 per cent.

16 To determine the carbon in the refuse the total refuse was spread out upon a large floor, broken up, thoroughly mixed and quartered. The quarter sample was ground, and this ground portion was in turn quartered to get the sample for chemical analysis. The calorific values of the coals were determined by the oxygen-bomb calorimeter.

17 The radiation and unaccounted for losses have been calculated for those tests, for which there were ultimate analyses of the coal. The tests were made during extremely cold weather, and there was over the boiler only a temporary building affording but little protection from the elements; the boiler room temperatures were so low at times the boiler pressure gage piping and the water column blow-off piping froze. In view of these facts and the drafty condition of the boiler room the radiation and unaccounted for losses of from 5 to 7 per cent are worthy of note.

18 The theoretical efficiency was calculated on the assumption that the only heat loss was that to the stack when the combus-

tion was complete without excess air, and the temperature of gas leaving the boiler was equal to the temperature of the wet steam in the boiler. The efficiencies in Table 1 have been calculated for those tests for which there were ultimate analyses of the coal.

19 It may be possible under favorable conditions to reduce the radiation losses to 2.5 per cent. With thorough mixing and a suitable type of combustion chamber the heat losses in the gas may be expected to be not more than 1.5 per cent greater than the theoretical gas loss; the maximum combined efficiency which may be attained under these most favorable conditions is about 85 per cent.

TABLE 1 THEORETICAL EFFICIENCIES

No. of Tests	Theoretical Efficiency, Per Cent	Combined Efficiency, Per Cent	Gain in Efficiency necessary to attain Theoretical Efficiency
8	90.10	79.60	10.50
11	90.15	78.14	12.01
16	90.05	76.18	13.87
18	89.95	75.20	14.75
25	89.52	75.79	13.73
35	89.90	78.33	11.57
39	89.85	78.65	11.20

SUMMARY OF RESULTS

20 The best combined efficiencies are obtained with 56 to 66 per cent of the boiler rating. The efficiencies fall off slowly as the quantity of steam generated is increased. The extreme variation in efficiency shown by the individual tests is from 75.2 to 81.3 per cent.

21 The gain in efficiency due to the superheater seems to bear no definite relation to the amount of moisture in the steam entering the superheater. By gain in efficiency due to superheater is meant the heat absorbed by the superheater in percentage of the total heat in the coal. The superheat shows a general tendency upward with the increase in boiler rating. For 60 per cent of the boiler rating the superheat is about 36 deg. fahr., and for 120 per cent rating about 74 deg. fahr. At 60 per cent rating the weight of steam passing through the super-

heat is 18,000 lb. per hour, the amount for which the superheater was built. The pressure drop through the superheater for its rated quantity of steam is 5 lb.; at 120 per cent boiler rating the weight of steam passing through the superheater is double that for which the superheater was designed, and the pressure drop through superheater is 28 lb.

22 At the low boiler ratings the last pass of the boiler, and in fact the last two passes, are of very little use. The best combined efficiencies were obtained when running between 56 and 66 per cent of boiler rating. At these low ratings the drop in gas temperature through the second and third passes altogether is about 20 deg. fahr., and the heat absorbed is very small, about 1.3 per cent of the total. These two passes have 50 per cent of the boiler heating surface. If the second and third passes were dropped off entirely, the loss in combined efficiency would be about 1 per cent at the low rating. It is only when the boiler has reached 75 per cent rating or more that the gain in economy in the last two passes is sufficient to give a reasonable return upon the investment in heating surface. While the exit gas temperatures were reduced almost to the lowest theoretical limit which can be reached without the use of an economizer, it was done at a large expenditure in heating surface.

23 When running at about 50 per cent of boiler rating the temperature of the gases leaving the boiler is practically that of the steam in the boiler. As the capacity is increased the difference between the temperature of the gases leaving the boiler and the temperature of the steam in the boiler increases. The increase in the difference between these two temperatures seems to be approximately in the same ratio as the increase in capacity, so that for an increase from 50 to 100 per cent of the boiler rating the difference in temperature between exit gases and steam increases about 50 deg. fahr.

24 An examination of the boiler after more than six months' service showed no evidence of pitting of the heating surface of the last pass due to the sulphur content of the coal and low exit gas temperature.

APPENDIX

RATING VS. COMBINED EFFICIENCY

25 Fig. 2, showing the relation between boiler rating and combined efficiency, indicates that the efficiency decreases slowly as the steam generated is increased. The line shows an efficiency of 80 per cent at 50 per cent of rating, and an efficiency of 73.5 per cent at 150 per cent rating, or a decrease of 6.5 per cent in the combined efficiency for an increase of 100 per cent in rating. This decrease in efficiency with an increase in rating seems to be due primarily to the temperature of the gases leaving the boiler.

EFFICIENCY AND RATING VS. TEMPERATURE OF GASES LEAVING BOILER

26 It may be seen from Fig. 2 that the temperature of the gases leaving the boiler goes up with the rating; the increase being about 1 deg. fahr. for 1 per cent rating. On the other hand an increase in the exit gas temperature gives a decrease in the efficiency. As an increase in exit gas temperature gives an increased rating and a decreased efficiency it follows that the efficiency will fall off as the rating increases.

27 The superheat-rating curve shows that the superheat has a general tendency upward with the rating.

28 The line for rating coal per retort per hour shows that the coal consumption increases at a more rapid rate than the steam generated.

GAS TEMPERATURES

29 The gas temperatures throughout the boiler from furnace to breeching are given in the tables, and the percentages of total heating surface (boiler plus superheater) between the several pyrometer positions are given on the chart, Fig. 3. While the heat transfer between the several pyrometer positions varies somewhat irregularly, the general tendency is to show a decrease in the percentage of heat absorption in the first pass with an increase in boiler rating. In the second and third passes the opposite condition prevails, the percentage of heat absorption increasing with boiler rating.

30 A study of the gas temperature data will explain these conditions. At the low boiler rating the gas temperatures are so low by the time the gases reach the second pass that the rate of absorption is low and the percentage small. In general the temperature of the gas entering the second pass increases with the boiler rating; this increases the rate of heat absorption, and gives to this portion of the boiler a greater proportion of the total work.

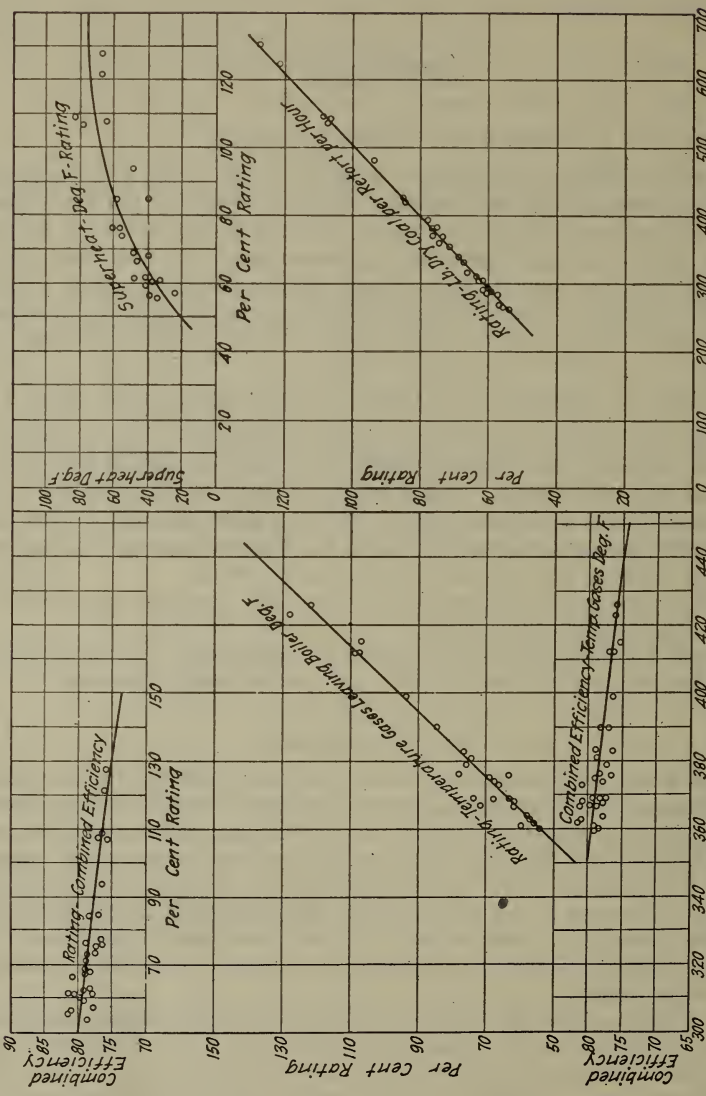

FIG. 2 CURVES SHOWING RELATION BETWEEN BOILER RATING AND COMBINED EFFICIENCY, TEMPERATURE OF GASES LEAVING BOILER, SUPERHEAT AND COAL PER RETORT HOUR

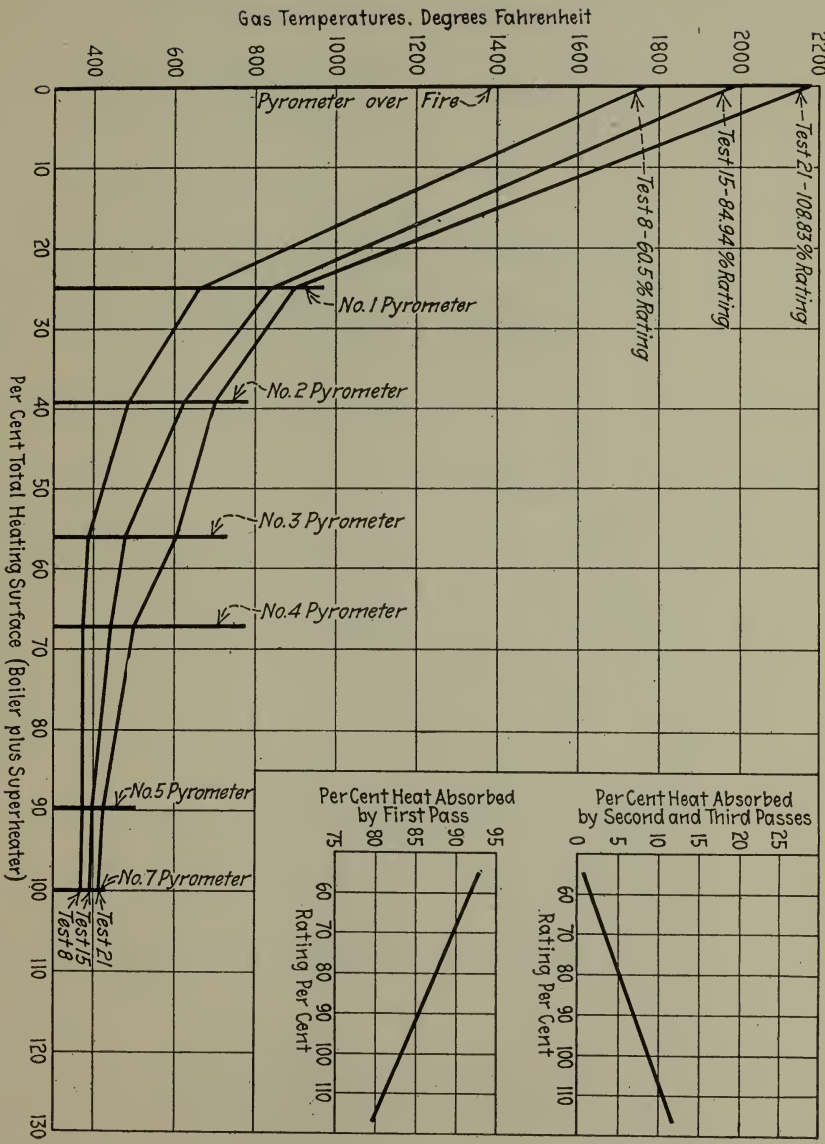

FIG. 3 TEMPERATURE OF GASES THROUGH BOILER

TABLE 2 1000-H.P. BOILER, TEST DATA

Test No.	Date of Test	Length of Test, Hours	Steam Pressure by Gage Lb. per Sq. In.		Barometer, Lb.	Average Draft, In. of Water			Temperature, Deg. Fahr						
			In Boiler	Leaving Superheater		Pressure in Stoker Wind Box	Suction over Fire	Suction before Breeching	Fireroom	Saturated Steam at Pressure, leaving Superheater	Steam leaving Superheater	Superheat	Feedwater entering Boiler	Gases leaving Boiler	Saturated Steam at Boiler Pressure
1	11-16-17-18-11	47.983	136.00	124.00	14.53	4.89	0.157	.....	41.9	352.26	392.00	39.74	147.92	390	358.77
3	11-21-22-11	24.28	127.00	124.00	14.57	2.480	0.148	.....	32.0	352.29	392.00	39.71	133.34	363	353.95
4	11-23-24-11	24.61	130.00	124.00	14.58	2.599	0.144	.....	44.6	352.27	393.80	41.53	142.34	368	355.59
5	11-24-25-11	21.61	126.00	121.00	14.57	2.599	0.146	.....	32.0	350.58	392.00	41.42	141.26	361	353.39
6	11-27-28-11	17.90	131.00	125.00	14.61	2.520	0.137	.....	48.2	352.85	401.00	48.15	146.12	368	356.11
7	11-28-29-11	21.50	130.00	125.00	14.54	2.599	0.137	.....	47.75	352.81	390.20	37.39	144.68	367	355.57
8	12-1-2-11	28.083	131.00	126.00	13.51	2.717	0.104	.....	38.66	354.81	387.50	32.69	142.16	367	355.55
9	12-4-5-11	24.33	126.00	123.00	14.75	2.402	0.098	.....	39.56	351.84	375.80	23.96	125.24	364	353.48
10	12-5-6-11	24.68	128.00	124.00	14.74	2.402	0.137	.....	40.64	352.36	386.60	34.24	125.60	362	354.58
11	12-6-7-11	21.66	141.00	128.00	14.74	3.228	0.126	.....	45.50	354.56	393.80	39.24	131.72	369	361.44
12	12-7-8-11	20.76	136.00	128.00	14.77	3.386	0.122	.....	46.04	354.58	401.00	46.42	129.74	373	358.90
13	12-8-9-11	24.30	136.00	127.00	14.89	3.662	0.137	.....	46.22	354.13	402.80	48.67	125.24	375	358.95
14	12-11-12-11	24.38	135.00	124.00	14.69	3.465	0.110	.....	59.72	352.33	408.20	55.87	145.04	381	358.30
15	12-12-13-11	23.76	134.00	126.00	14.63	4.370	0.118	.....	43.16	353.41	411.80	58.39	139.64	390	357.73

TABLE 2—*Continued*

16	12-13-14-11	18.68	135.00	129.00	14.71	3.110	0.098	.....	39.56	355.12	411.80	56.68	137.48	383	358.32
17	12-14-15-11	23.93	134.00	127.00	14.69	3.465	0.091	0.196	40.10	354.00	415.40	61.40	157.46	383	357.42
18	12-19-20-11	29.11	137.00	126.00	14.84	4.567	0.118	0.359	34.88	353.52	410.00	56.48	137.66	401	359.42
19	12-21-22-11	24.11	140.00	127.00	14.72	5.276	0.137	0.394	36.50	354.02	408.20	54.18	130.64	396	360.86
21	12-26-27-11	22.13	147.00	130.00	14.30	5.040	0.085	0.472	43.88	355.44	438.80	83.36	131.18	412	364.24
22	12-27-28-11	23.70	143.60	127.00	14.42	4.764	0.098	0.484	36.50	353.84	433.40	79.56	119.84	415	362.70
23	12-28-29-11	20.70	144.00	127.00	14.65	4.685	0.173	0.496	40.64	353.98	402.80	48.82	125.42	399	362.93
24	12-29-30-11	23.55	145.00	128.00	14.78	4.488	0.122	0.500	37.94	354.58	419.00	64.42	145.22	412	363.49
25	1-4-5-12	18.00	158.00	129.00	14.53	5.630	0.085	0.461	25.16	355.01	422.60	67.59	141.80	423	369.66
26	1-5-6-12	23.96	150.00	123.00	14.61	5.906	0.075	0.512	12.20	351.77	419.00	67.23	140.90	426	365.85
27	1-18-19-12	22.28	141.00	134.00	14.56	3.071	0.158	0.276	43.70	357.67	404.60	46.93	134.06	379	361.35
29	1-22-23-12	23.95	138.00	129.00	14.57	3.110	0.110	0.173	29.30	355.05	388.40	33.35	139.10	367	359.80
30	1-23-24-12	22.36	137.00	129.00	14.49	3.189	0.106	0.268	31.64	354.99	395.60	40.61	139.10	369	359.26
33	1-26-27-12	24.15	139.00	131.00	14.58	3.583	0.126	0.389	12.56	356.09	392.00	35.91	138.92	376	360.28
34	1-29-30-12	24.016	138.00	132.00	14.53	2.717	0.110	0.358	29.48	356.61	392.00	35.39	146.30	374	359.76
35	1-30-31-12	20.40	134.00	128.50	14.46	2.638	0.126	0.354	29.30	354.69	381.20	26.51	157.28	376	357.64
37	2-1-2-12	25.26	134.00	128.00	14.56	2.795	0.110	0.252	22.46	354.47	377.60	23.13	152.24	369	357.67
39	2-6-7-12	24.53	132.00	127.00	14.60	2.126	0.079	0.126	26.24	353.95	377.60	23.65	157.28	360	356.67
40	2-7-8-12	18.28	135.00	130.00	14.55	2.205	0.067	0.078	26.42	355.57	368.60	13.03	160.70	360	358.23

Tests 1-19 inclusive with mixed coals from mines Nos. 24, 25, 27 and 29.
Tests 21-40 inclusive with coal from No. 29 mine only.

TABLE 3 1000-H.P. BOILER, TEST RESULTS

Test No.	Relative Humidity of Atmosphere, per Cent	Feedwater, Lb. per Hr.	Evaporation Factor Combined Unit	Evaporation Factor Boiler only	Lb. Dry Coal per Hr.	Lb. Dry Refuse per Hr.	Lb. Combustible ascending from Grate per Hr.	Lb. Dry Coal per Hr. per Sq. Ft. Grate.	Lb. Dry Coal per Hr. per Retort	Equivalent Evaporation per Lb. Dry Coal Combined Unit	Equivalent Evaporation per Lb. Dry Coal Boiler only	Per Cent Builders Rating Developed	Efficiencies			Gain in Efficiency Due Superheater
													Boiler, Furnace, and Superheater	Boiler and Furnace	Boiler Only	
1	....	25814	1.13317	1.09201	2538	263	2275	40.63	423.00	11.525	11.106	84.79	78.02	75.18	75.20	2.84
3	....	17045	1.14817	1.10516	1615	144	1471	25.85	269.17	12.121	11.666	56.72	80.89	77.85	77.54	3.04
4	....	18692	1.13998	1.09712	1828	150	1678	29.26	304.67	11.658	11.219	61.77	77.93	74.99	74.62	2.94
5	....	17994	1.14048	1.09829	1736	158	1579	27.79	289.33	11.818	11.381	59.48	79.10	76.18	76.25	2.92
6	....	18590	1.14004	1.09539	1756	154	1603	28.11	292.67	12.067	11.594	61.43	80.76	77.59	77.43	3.17
7	....	18487	1.13592	1.09710	1732	138	1594	27.73	288.67	12.125	11.711	60.87	81.00	78.23	77.43	2.77
8	....	18400	1.13632	1.09888	1759	143	1616	28.16	293.17	11.889	11.498	60.60	79.60	76.99	76.30	2.61
9	....	17253	1.14706	1.11486	1711	138	1573	27.39	285.17	11.567	11.243	57.36	77.71	75.53	74.71	2.18
10	....	16719	1.15298	1.114427	1610	158	1453	25.77	268.33	11.972	11.752	55.88	81.30	79.81	78.17	1.49
11	....	20366	1.15026	1.10953	1989	158	1831	31.84	331.50	11.775	11.358	67.94	78.14	75.37	75.22	2.77
12	....	19817	1.15645	1.11249	1907	184	1723	30.53	317.83	12.018	11.561	66.43	80.69	77.61	78.05	3.08
13	....	20440	1.16219	1.11622	2030	206	1824	32.50	338.30	11.704	11.241	68.86	78.89	75.77	76.23	3.12
14	....	22365	1.14537	1.09763	2165	181	1984	34.66	360.80	11.833	11.339	74.25	78.64	75.36	75.40	3.28
15	....	25424	1.15264	1.10235	2557	246	2311	40.93	426.20	11.461	10.961	84.94	76.88	73.52	74.16	3.36

TABLE 3—*Continued*

16	....	22776	1.15445	1.10462	2296	209	2087	36.76	382.66	11.453	10.958	76.21	76.18	72.88	73.47	3.30
17	....	23311	1.13620	1.08445	2233	189	2044	35.75	372.17	11.864	11.323	76.77	78.76	75.17	75.67	3.59
18	....	26640	1.15365	1.10354	2708	243	2465	43.35	451.33	11.349	10.857	89.08	75.20	71.94	72.45	3.26
19	53.0	26997	1.15974	1.11113	2784	261	2523	44.57	464.00	11.247	10.776	90.75	75.42	72.26	73.20	3.16
21	53.5	31933	1.17576	1.11153	3293	339	2954	52.72	548.80	11.401	10.778	108.83	76.21	72.04	73.50	4.17
22	49.5	31128	1.18480	1.12276	3278	333	2945	52.48	546.30	11.252	10.663	106.90	75.50	71.55	72.50	3.95
23	55.1	27889	1.16200	1.11551	2894	315	2579	46.33	482.30	11.197	10.749	93.93	76.18	73.13	73.70	3.05
24	55.3	32282	1.15060	1.09658	3229	318	2911	51.69	538.20	11.504	10.964	107.66	76.87	73.26	74.29	3.61
25	51.3	38110	1.15605	1.10142	3913	443	3470	62.64	652.20	11.258	10.726	127.70	75.79	72.21	74.02	3.58
26	53.4	36225	1.15576	1.10538	3744	452	3292	59.94	624.00	11.181	10.694	121.36	75.93	72.62	74.23	3.31
27	51.0	22631	1.15327	1.10808	2294	234	2060	36.72	382.30	11.375	10.929	75.65	77.28	74.25	74.36	3.03
29	53.0	21690	1.13932	1.10299	2136	224	1912	34.20	356.00	11.569	11.200	71.63	78.76	76.25	76.43	2.51
30	51.9	22249	1.14357	1.10299	2208	215	1993	35.35	368.00	11.523	11.114	73.75	77.49	74.74	74.94	2.75
33	51.7	23520	1.14127	1.10314	2369	237	2131	37.93	394.80	11.333	10.954	77.80	76.48	73.92	74.37	2.56
34	54.0	20553	1.13357	1.09515	1996	183	1813	31.95	332.70	11.670	11.275	67.53	78.77	76.11	75.71	2.66
35	53.0	19485	1.11639	1.08318	1864	157	1707	29.84	310.70	11.668	11.321	63.05	78.33	76.00	75.55	2.33
37	52.3	19249	1.11952	1.08892	1835	162	1673	29.38	305.80	11.742	11.421	62.46	79.08	76.92	76.61	2.16
39	49.4	16710	1.11448	1.08081	1583	124	1459	25.34	263.80	11.764	11.409	53.98	78.65	76.27	75.66	2.38
40	50.0	16775	1.10487	1.07883	1587	138	1449	25.41	264.50	11.675	11.400	53.72	78.79	76.94	76.33	1.85

Tests 1–19 inclusive with mixed coals from mines No. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

TABLE 4 1000-H.P. BOILER, TEMPERATURE AND DRAFTS

Test No.	Furnace	Gas Temperatures, Deg. Fahr.							Draft in Stoker and Wind Boxes, In. Water						
		No. 1 In First Pass over 14th Tube	No. 2 In First Pass over 24th Tube	No. 3 In Second Pass over 24th Tube	No. 4 In Second Pass over 14th Tube	No. 5 In Third Pass over 14th Tube	No. 6 In Third Pass over 24th Tube	No. 7 Leaving Boiler between Boiler and Back Wall)	East Tuyere Box	West Tuyere Box	Bottom West Outside Box	Bottom East Outside Box	Center or Main Wind Box	Top East Outside Box	Top West Outside Box
1	....	759	572	460	423	388	385	390	4.01	4.13	4.76	5.19	4.89	6.14	5.98
3	....	691	518	387	399	378	378	363	2.16	2.20	2.87	2.95	2.48	4.09	4.01
4	....	631	498	406	361	365	363	368	2.32	2.40	2.91	3.18	2.59	4.05	3.89
5	....	639	509	406	365	367	370	361	2.32	2.40	2.91	3.11	2.59	4.05	3.97
6	....	666	500	408	390	378	367	368	2.32	2.32	2.91	3.07	2.51	4.01	3.97
7	....	639	482	387	381	370	363	367	2.36	2.36	2.95	3.11	2.59	3.97	3.93
8	1763	658	487	385	370	370	369	367	2.44	2.51	2.99	3.14	2.71	4.05	3.97
9	1841	626	417	365	352	356	363	364	2.16	2.24	2.79	2.87	2.40	3.89	3.81
10	1800	644	493	379	365	363	370	362	2.24	2.32	2.83	2.87	2.40	3.93	3.85
11	1888	696	505	387	383	363	372	369	2.95	3.07	3.50	3.66	3.22	4.60	4.52
12	1866	738	532	396	392	370	379	373	3.18	3.22	3.66	3.81	3.38	4.68	4.60
13	1958	801	572	428	414	385	390	375	3.42	3.50	3.85	4.09	3.66	4.68	4.60
14	1902	738	554	424	408	379	381	381	3.18	3.22	3.66	3.97	3.46	4.60	4.52
15	1989	838	621	477	441	396	401	390	4.05	4.17	4.76	5.07	4.37	6.10	5.94

TABLE 4—*Continued*

16	....	727	538	423	390	378	374	383	2.83	2.95	3.50	3.74	3.11	4.68	4.60
17	....	788	594	433	408	385	385	383	3.14	3.30	3.70	3.97	3.46	4.68	4.60
18	....	851	651	496	394	405	403	401	4.13	4.29	4.76	5.19	4.56	5.90	5.82
19	....	810	612	518	441	401	397	396	4.80	5.00	5.35	5.70	5.27	6.29	6.18
21	2176	900	698	603	495	426	419	412	4.52	4.68	5.07	5.51	5.03	6.10	6.02
22	....	914	712	604	491	424	414	415	4.17	4.33	4.84	5.39	4.76	6.02	5.90
23	....	838	617	507	459	401	397	399	4.13	4.21	4.56	4.96	4.68	5.51	5.39
24	....	867	653	525	468	412	405	412	3.93	4.09	4.48	4.92	4.48	5.51	5.39
25	....	961	711	541	491	426	423	423	4.92	5.00	5.59	6.10	5.63	6.85	6.73
26	....	982	752	563	504	438	437	426	5.35	5.43	5.98	6.45	5.90	7.08	6.88
27	....	819	610	478	439	401	387	379	2.71	2.83	3.03	3.42	3.07	3.62	3.62
29	....	689	570	423	401	385	383	367	2.71	2.87	3.18	3.66	3.11	4.05	3.97
30	....	703	612	471	435	396	396	369	2.83	2.91	3.18	3.58	3.18	3.81	3.74
33	....	792	599	464	428	399	388	376	3.18	3.26	3.54	3.93	3.58	4.25	4.21
34	....	716	550	455	414	381	379	374	2.44	2.55	2.71	2.95	2.71	3.14	3.11
35	....	714	549	453	408	381	378	376	2.28	2.40	2.59	2.83	2.63	3.11	3.07
37	....	727	545	442	405	379	377	369	2.48	2.51	2.75	2.99	2.79	3.18	3.18
39	....	649	496	397	388	367	363	360	1.92	2.04	2.24	2.48	2.12	2.83	2.79
40	....	603	446	370	367	352	354	360	1.96	2.12	2.24	2.40	2.20	2.63	2.63

Tests 1–19 inclusive with mixed coals from mines No. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

TABLE 5 1000-H.P. BOILER, GAS ANALYSIS

Test No.	Analysis of Gas in First Pass. Top of 14th Tube			Analysis of Gas before Breeching. Gas Leaving Boiler		
	Per Cent CO_2	Per Cent 0	Per Cent CO	Per Cent CO_2	Per Cent 0	Per Cent CO
1	11.70	5.42	0.160	10.32	7.44	0.035
3	10.70	6.60	0.166	No record		
4	10.22	5.35	0.040	10.10	Machine record	
5	10.47	5.83	0.070	9.39	Machine record	
6	10.15	7.25	0.050	9.00	Machine record	
7	10.30	7.53	0.000	9.00	Machine record	
8	10.64	7.54	0.064	10.38	8.17	0.030
9	10.62	7.88	0.000	10.16	8.41	0.025
10	11.08	7.85	0.058	10.02	8.60	0.000
11	10.90	7.66	0.030	10.30	Machine	
12	12.13	6.37	0.040	11.20	Machine	
13	12.28	4.98	0.250	10.50	Machine	
14	12.09	6.34	0.070	10.59	7.97	0.012
15	12.21	6.05	0.020	10.44	8.00	0.002
16	11.75	6.42	0.000	10.54	8.06	0.057
17	12.02	6.29	0.100	10.68	7.73	0.046
18	11.42	7.40	0.020	10.22	8.52	0.004
19	12.19	5.94	0.067	10.77	7.86	0.022
21	11.89	6.69	0.000	10.75	8.22	0.000
22	12.59	6.13	0.071	10.89	7.87	0.059
23	11.40	7.31	0.057	10.15	8.44	0.029
24	11.91	6.61	0.022	10.00	8.47	0.000
25	13.70	4.30	0.016	11.66	6.03	0.0098
26	13.35	4.60	0.025	11.35	6.78	0.000
27	11.67	7.00	0.000	10.17	8.30	0.000
29	11.07	7.63	0.000	10.01	8.65	0.000
30	11.43	7.07	0.033	9.84	8.83	0.011
33	10.12	8.68	0.000	9.24	9.56	0.000
34	9.91	8.99	0.090	9.31	9.44	0.026
35	10.81	7.84	0.000	9.39	9.45	0.000
37	10.60	8.12	0.000	9.42	9.15	0.000
39	10.39	8.22	0.000	8.84	9.83	0.000
40	10.69	7.96	0.000	9.43	9.43	0.000

Tests 1–19 inclusive with mixed coals from mines No. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

TABLE 6 1000-H.P. BOILER. COAL ANALYSIS. PERCENTAGES BY WEIGHT CALORIFIC VALUES

Test No.	Proximate Analysis Dry Coal			Moisture in Coal as Fired	Carbon in Dry Refuse, per Cent	B.t.u. per Lb. Dry Coal	B.t.u. per Lb. Combustible.	Per Cent Sulphur in Dry Coal
	Volatile Matter	Fixed Carbon	Ash					
1	28.99	62.00	9.01	2.88	15.49	14334	15987	2.59
3	29.50	62.53	7.97	3.20	10.30	14541	16031	2.94
4	29.97	62.53	7.50	3.26	10.09	14517	15894	2.40
5	29.74	62.51	7.75	3.04	13.91	14498	15926	2.62
6	28.18	63.90	7.92	2.97	8.64	14499	15922	1.83
7	28.86	63.31	7.83	3.78	1.44	14526	15945	1.92
8	28.85	63.39	7.76	4.29	3.90	14493	15913	2.49
9	29.22	62.97	7.81	2.97	3.39	14445	15881	2.57
10	29.24	61.76	9.00	2.60	9.69	14289	15922	2.57
11	29.63	63.29	7.08	2.95	10.93	14624	15922	2.44
12	29.22	62.76	8.02	2.97	17.91	14454	15910	2.24
13	29.70	62.03	8.27	3.29	18.08	14396	15922	3.03
14	29.78	62.98	7.24	2.80	14.75	14602	15922	2.20
15	29.98	62.35	7.67	3.11	23.79	14467	15869	2.32
16	29.79	63.00	7.21	3.41	23.52	14590	15922	2.15
17	30.20	63.00	6.80	3.74	21.10	14618	15860	2.07
18	29.42	63.37	7.21	3.40	20.05	14645	15971	2.29
19	30.57	62.34	7.09	3.15	24.50	14471	15761	2.31
21	28.40	64.13	7.47	3.60	28.73	14518	15864	1.77
22	28.36	63.75	7.89	4.10	21.48	14462	15883	1.83
23	28.31	62.65	9.04	4.90	17.80	14263	15885	1.95
24	28.40	64.05	7.55	3.99	25.76	14522	15883	1.90
25	28.15	63.76	8.09	3.77	30.68	14415	15858	1.56
26	28.30	62.70	9.00	4.15	29.18	14290	15903	1.83
27	27.86	63.20	8.94	4.27	14.94	14284	15883	1.88
29	28.12	62.78	9.10	4.73	14.37	14254	15885	2.06
30	28.20	63.45	8.35	3.76	16.23	14430	15947	2.18
33	28.56	63.06	8.38	3.70	20.14	14380	15883	1.94
34	28.27	63.21	8.52	4.30	7.99	14376	15915	2.08
35	28.66	63.40	7.94	4.68	7.07	14455	15883	1.91
37	28.11	63.73	8.16	4.61	7.59	14409	15871	1.79
39	28.92	63.57	7.51	4.12	4.69	14515	15883	2.16
40	29.27	62.51	8.22	4.23	5.09	14378	15874	2.35

Tests 1–19 inclusive with mixed coals from mines No. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

TABLE 7 1000-H.P. BOILER, RESUMÉ OF PRINCIPAL RESULTS

Test No.	Per Cent Rating	Combined Efficiency	Temperature of Gases leaving Boiler, Deg. Fahr.	CO_2 in Gases leaving Boiler, Per Cent	Per Cent Combustible in Dry Refuse	B.t.u. per Lb. Dry Coal	Evaporation from and at 212 Deg. Fahr. per Lb. Dry Coal Combined Unit	Quality of Steam leaving Boiler	Radiation and Unaccounted for Losses
1	84.79	78.02	390	10.32	15.49	14334	11.525	97.92	....
3	56.72	80.89	363	.....	10.30	14541	12.121	97.83	....
4	61.77	77.93	368	10.10	10.09	14517	11.658	97.93	....
5	59.48	79.10	361	9.39	13.91	14498	11.818	97.99	....
6	61.43	80.76	368	9.00	8.64	14499	12.067	98.16	....
7	60.87	81.00	367	9.00	1.44	14526	12.125	98.20	....
8	60.60	79.60	367	10.38	3.90	14493	11.889	98.11	5.46
9	57.36	77.71	364	10.16	3.39	14445	11.567	98.00	....
10	55.88	81.30	362	10.02	9.69	14289	11.972	97.97	....
11	67.94	78.14	369	10.30	10.93	14624	11.775	97.94	6.42
12	66.43	80.69	373	11.20	17.91	14454	12.018	98.11	....
13	68.86	78.89	375	10.50	18.08	14396	11.704	98.01	....
14	74.25	78.64	381	10.59	14.75	14602	11.833	98.23	....
15	84.94	76.88	390	10.44	23.79	14467	11.461	98.15	....
16	76.21	76.18	383	10.54	23.52	14590	11.453	98.14	6.99
17	76.77	78.76	383	10.68	21.10	14618	11.864	98.20	....
18	89.08	75.20	401	10.22	20.05	14645	11.349	98.00	7.15
19	90.75	75.42	396	10.77	24.50	14471	11.247	98.01	....
21	108.83	76.21	412	10.75	28.73	14518	11.401	98.03	....
22	106.90	75.50	415	10.89	21.48	14462	11.252	98.02	....
23	93.93	76.18	399	10.15	17.80	14263	11.197	97.84	....
24	107.66	76.87	412	10.00	25.76	14522	11.504	98.00	....
25	127.70	75.79	423	11.66	30.68	14415	11.258	97.99	5.65
26	121.36	75.93	426	11.35	29.18	14290	11.181	98.42	....
27	75.65	77.28	379	10.17	14.94	14284	11.375	98.05	....
29	71.63	78.76	367	10.01	14.37	14254	11.569	98.10	....
30	73.75	77.49	369	9.84	16.23	14430	11.523	98.11	....
33	77.80	76.48	376	9.24	20.14	14380	11.333	98.09	....
34	67.53	78.77	374	9.31	7.99	14376	11.670	98.05	....
35	63.05	78.33	376	9.39	7.07	14455	11.668	98.04	5.00
37	62.46	79.08	369	9.42	7.59	14409	11.742	98.10	....
39	53.98	78.65	360	8.84	4.69	14515	11.764	97.80	4.75
40	53.72	78.79	360	9.43	5.09	14378	11.675	97.93	....

Tests 1–19 inclusive with mixed coal from mines Nos. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

DISCUSSION

W. D. Ennis. The results of these tests do not apparently harmonize with principles usually accepted. It is customary to think of a 3 or 3½ lb. rate of evaporation as that of maximum efficiency. At rates below this, the efficiency is reduced by the increased proportion of radiation loss, and by the lower furnace efficiency, resulting from the low rate of combustion and necessarily excessive air supply. At evaporation rates higher than 3 or 3½ lb., efficiency is gradually reduced because of increased loss to the stack, as indicated by rising flue gas temperatures.

In this boiler, the efficiency steadily decreased as the rating increased, and the best efficiency was reached at an evaporation rate below 2 lb. When efficiencies are said to decrease at low rates the usual proportions of design are assumed: say with soft coal, a heating surface to grate surface ratio around 50. A 3 lb. evaporation rate means with this ratio, a rate of combustion of about 15 lb. per sq. ft. of grate per hour. No rate of combustion as low as this was reached in the present tests. In this boiler, with its unusual ratio of heating to grate surface, low rates of evaporation were not accompanied by low rates of combustion. At no time was the furnace efficiency notably lowered from the good average maintained.

It may seem that this scarcely explains the rate at which the efficiency apparently decreases with increased evaporation. As large a heating surface as was provided should have more completely absorbed the furnace heat. If Fig. 1-a is a scale drawing, a slightly better distribution of gas velocities would have been possible. It is quite easy, however, to over-emphasize the rate of decrease of efficiency, particularly as the *rating-combined efficiency* and *combined efficiency-gas temperature* curves of Fig. 2 are both represented to a distorted scale. The base line is that of 70 per cent efficiency, not of zero efficiency. The man on the street would say that the efficiency was about constant. In fact the curve of coal consumption against percentage of rating, Fig. 2, can scarcely be distinguished from a straight line. The flue gas temperature was 360 deg. at 56 per cent of rating and 400 deg. at 96 per cent: both figures are unusually good, with not much difference between them, certainly not more than would have been offset by the probably excessive air supply at the lower temperature and rating had the boiler had a grate of normal size.

The difference between the efficiencies at half load and full load was about 1½ per cent. This means an excess coal consumption at full load of about 0.05 lb. per h.p-hr. Under the most unfavorable conditions, continuous service, the boiler running at full load would waste about 1/5 ton of coal per year per horsepower capacity. To offset this waste it would be necessary to duplicate boiler, piping, smoke flue, etc., at an initial cost of perhaps $20 per h.p. The saving of 1/5 ton of coal per year, which is the maximum possible, would usually not pay fixed charges on an investment of $20.

William Kent. Fig. 2 confirms what has been generally found, that there is a slight decrease in efficiency with increased rate of driving of the boiler, and that decrease will be less the more the theoretical ratio of air approaches the combustible. If air supply is excessive the efficiency will fall off much more rapidly than is shown in Fig. 2. The reason why the maximum efficiency is shown at such a low rate of driving is undoubtedly due to the fact that this boiler was thoroughly covered with nonconducting material and the loss by radiation was extremely slight.

Professor Ennis has spoken of the rate of combustion and the ratio of heating to grate surface. Probably 15 years ago, at a meeting of this Society, in discussing a paper by W. W. Christie, the writer showed that there was no change in efficiency necessarily due to a change in the rate of combustion, between 5 lb. and 240 lb. per sq. ft. of grate surface per hour, except the loss due to sparks and cinders blown into the stack at the higher rate of driving. There is no necessary relation either between the rate of combustion and the excess air supply over that theoretically required; that is, with a rate of combustion as low as 5 lb. per sq. ft. of grate surface per hour, perfect combustion is possible with a certain small excess air supply, and at 20 times this rate without any greater percentage of excess air. The two variables are the thickness of fire and the amount of draft, both of which are under control of the fireman, and excellent results are also attainable with a high rate of combustion, provided the grate surface is small enough, and the draft and thickness of fire regulated to get the proper analysis of the gases. If there is about 3 per cent of oxygen in the waste gases, and in the case of a coal-burning furnace, an even fire, there will not be more than

one-half of 1 per cent of CO, and that condition will give the maximum boiler efficiency, provided the baffles are properly arranged and there are no leaks of air or other troubles, and the boiler is not being driven beyond its most economical rate.

J. H. Browne said that the method of baffling this boiler was somewhat new. He had seen trials with a similar method of baffling, but the experiments were not continued long enough to be conclusive. It seemed to him that although the method of developing a baffling material was interesting, the creeping of the tubes would break up the material, and he asked if the author had data on this.

The interesting fact was shown that the best economy was at from 50 to 60 per cent of the boiler rating, possibly accounted for by the superior installation of the boiler. In view of the fact that the ratio of 56 per cent appeared to be the best, this large boiler had at least one advantage from a commercial standpoint. Anyone who had to do with boiler-room operation no doubt knew that the average fireman liked to run about twice as many boilers as necessary. With a plant designed so that it would run at best efficiency at 56 per cent, this matter had been helped. On the other hand, a good deal more money was invested for the same size boiler under these conditions.

D. S. Jacobus. Mr. Kent has said that the percentage of oxygen in the flue gases might to advantage be made lower than is indicated by the tests. An important feature of the analyses is that they nearly all show some carbon monoxides. The determination of carbon monoxide in boiler tests is often subject to considerable error. In the present instance the analyses were undoubtedly correct, and the limitations in efficiency through striving to run with but little excess air, or a low percentage of oxygen in the flue gases, is clearly brought out.

On decreasing the amount of excess air there is a tendency to approach a zone where carbon monoxide will appear in the flue gases. The better the furnace the higher the percentage of carbon dioxide will be before the carbon monoxide appears. In endeavoring to obtain high efficiencies, we take great care to watch for carbon monoxide. It is often an easy matter to get, say, 10 or 11 per cent of carbon dioxide with little or no carbon monoxide, but after exceeding 13 per cent of carbon dioxide the fur-

nace must be exceptionally good to consume all of the carbon monoxide.

Measuring the furnace temperatures has been suggested. It is extremely difficult, if not impossible, to do with accuracy, as the effect of the radiant heat complicates matters. A long time ago I calibrated a Uehling-Steinbart pyrometer, and great difculty was experienced in obtaining comparative temperatures. In an ordinary furnace the temperature of the gases may be different from the temperature due to the radiant heat and on this account there are variations depending on the form of the end of the pyrometer which is exposed to the conducted and the radiant heat. The arrangement of the water heating surface with respect to the furnace also affects the temperature measurements, and the same applies in a lesser degree to temperature determinations of the gases at different points within the setting.

C. D. Young. In some recent work with superheater locomotives we have found that generally the superheat will increase with increased evaporation or increased burning rate of the boiler. This is confirmed in stationary practice by the conclusions drawn in Par. 21. Fig. 2 which illustrated this, however, has the line curved in the upper right hand corner, whereas our tests with a superheater applied to water tube locomotive boiler seem to indicate that the line should be straight; that is, the increase in superheat is almost directly proportional to the increase in the burning rate.

A question which we have before us just now in locomotive practice is the determination of the effectiveness of water and superheating heating surface in a boiler and I would like to know if Mr. Bump has data to show what the heat transference rates are as between the flue gases and the water and the flue gases and the steam. Our results seem to indicate that the superheater heating surface *per se* is only about half as effective as the water heating surface in heat transfer.

The Author. Professor Ennis has made the statement that "as large a heating surface as was provided should have more completely absorbed the furnace heat." I would call his attention to the last two columns of Table 3 which show that there was very little difference between the temperatures of the gases leaving the boiler and the saturated steam in the boiler. The temperature of the fluid within the tubes is the minimum limit

of the exit gas temperature, and no amount of heating surface will reduce this limit.

The difference between the efficiencies at half load and full load was more than 3 per cent, not about 1½ per cent as Professor Ennis has stated. This means that the boiler running at full load would waste 0.5 tons of coal per horsepower-year, instead of the 1/5 ton stated in the discussion.

Although there is fuel economy in running at half load, yet fuel economy is not all there is to the cost of steam.

These tests are not put forward to show that the cheapest steam is to be obtained at low boiler rating. This would only be true with a very high priced fuel. With the usual prices of coal, and interest and depreciation taken into account, this boiler would need to run at 150 per cent rating at least, in order to generate steam at the lowest cost.

In reply to Mr. Young relative to the superheat curve of Fig. 2, the points obtained by these tests as well as from a later series of tests made after the baffles had been removed indicate that the line should be a curve, not a straight line. In other words, the superheat did not increase as fast as the boiler rating. The fact that the superheater was too small for the quantity of steam generated at the highest ratings may have had some influence upon the superheat.

In regard to the relative rates of heat transfer in the superheater and in the boiler, we have no data that will give an absolute measure of this transfer, but can get an approximate comparison from the drop in temperature of the gases as they pass over the several surfaces. Making calculations on this basis, we find that the rate of heat transfer for the superheater is fully as great as for the adjacent portions of the boiler tubing.

No. 1363

AIR IN SURFACE CONDENSATION

By Geo. A Orrok, New York

Member of the Society

In the paper on The Transmission of Heat in Surface Condensation[1], the subject of the effect of air on vacuum and heat transference was considered and reference was made to the few authorities who had investigated this subject. The author was unable to find that any one had quantitatively measured the air in a condenser, except J. A. Smith who had introduced known quantities of air and noted the effect on vacuum and heat transference.

2 In view of the lack of knowledge of this subject the author determined to investigate:

a The amounts of air contained in feedwater and those entering the condenser through the prime mover.

b The amounts of air removed from the condenser by means of the air pump.

c The effect of known quantities of air on the action of a surface condenser of commercial size.

3 The problem of air in the feedwater has been investigated by Smith whose paper appeared in London Engineering, October 7, 1904. He gives a table showing that the gases present in various samples of fresh and salt water varied from approximately 7/10 of one per cent to as much as 3½ per cent by volume. He also investigated the emission of gas from water when heated to about the boiling point and gave figures and curves showing his results.

4 The author's investigations were made at the Waterside No. 2 station of the New York Edison Company and the water samples were taken both from the suction and discharge sides

[1] Trans. Am.Soc.M.E., vol. 32, p. 1139.

Presented at the Annual Meeting 1912, of The American Society of Mechanical Engineers.

of the feed pump. The average temperature of the feedwater during the tests, after having passed the heater, was about 190 deg. fahr. It was thought possible that there might be air bubbles mechanically entrained in the water, besides the amount of air in solution. To determine the amount of this entrained air samples of the water were allowed to flow through the apparatus which is shown in Fig. 1. The water entered through

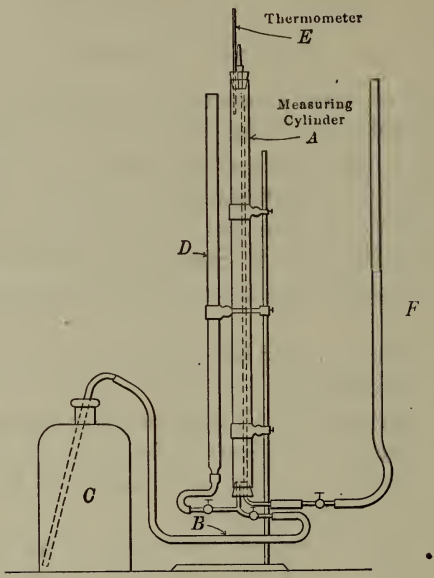

FIG. 1 APPARATUS USED TO DETERMINE AMOUNT OF AIR MECHANICALLY ENTRAINED IN WATER

the tube *F* and passed to the top of the measuring cylinder where the air separated and was collected at the top of the cylinder; the water flowed out of the cylinder through the tube *B* into the glass demijohn where it was measured. These tests were started with the apparatus entirely filled with water. The air collected was measured at atmospheric pressure by means of the pressure-regulating cylinder *D*. The temperature of the air was determined by the thermometer *E*. The amount of air dissolved in the feedwater was determined chemically by means of the following reactions:

$$2Mn(OH)_2+O_2=2H_2MnO_3$$
$$2H_2MnO_3+8HCl=2MnCl_2+6H_2O+2Cl_2$$
$$4KI+2Cl_2=4KCl+2I_2$$

5 The free iodine liberated was titrated with a solution of sodium thiosulphate of known strength and its amount determined. From this amount the quantity of oxygen in solution in the sample of water taken was calculated. The amount of air in solution was then figured from the known ratios of oxygen and nitrogen in solution in water at various temperatures. The determination in this way may be considered quite accurate.

6 This determination of air in water was also made of a number of samples of Croton water (the make-up feed) and from hot-well water. The following results, in per cent, were obtained of air by volume at atmospheric pressure:

	187 deg. fahr.
Mechanically entrained in feedwater, average of ten tests	0.0151
In solution in feedwater, average of ten tests	0.916
Total air in feedwater from open heaters	0.931

	52 deg. fahr.
In solution in Croton water, average of three samples	4.325

	80 deg. fahr.
In solution in hot-well water, average of four tests	0.269

7 These results show that Croton water gives off most of its dissolved air in passing through the open heaters, the amount of air occluded in the feedwater (0.931 per cent) being comparatively small. At 81 deg. fahr. and 2 in. of mercury absolute pressure when saturated with water vapor, this amount of air should be between 20 and 30 per cent of the feedwater volume.

8 The results obtained accord well with reports from other sources, although the author could find but little quantitative information on the subject. The fact was established, however, that very little air came into the condenser from the feedwater.

9 Having obtained the portions of air which enter the prime mover along with the steam, the next object of investigation was to determine the amount of air abstracted from the condenser by the air pump. Here again there were no precedents but it was determined to catch the air over water in a gasometer. For this purpose the discharge pipe of the air pump was piped to a

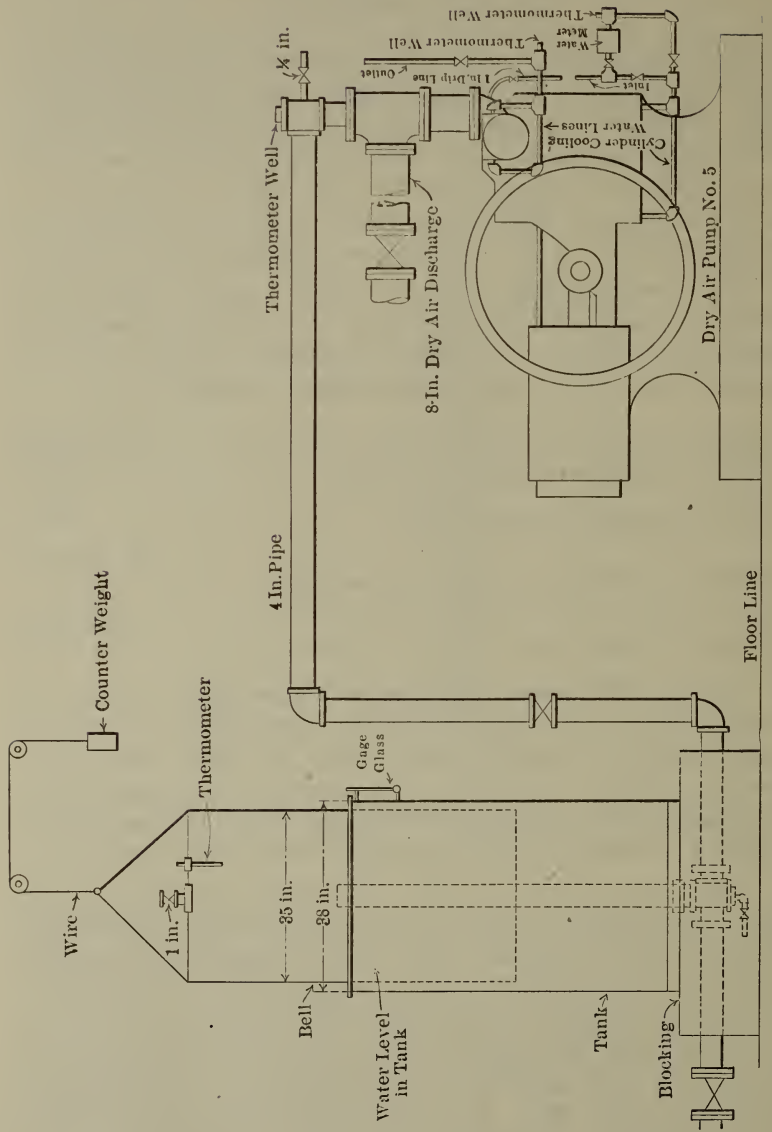

FIG. 2 LAYOUT FOR MEASURING AIR DISCHARGED BY DRY VACUUM PUMP

gasometer bell located in the basement in the vicinity of the air pump. A sketch plan of this outfit is shown in Fig. 2.

10 After the air pump had been running for a sufficient time to make sure that the conditions were constant, the valve was opened and the discharge from the air pump was allowed to enter the bell allowing it to rise. The time required to fill the bell was noted with a stop watch and from these data the amount of air delivered was calculated. The air in the bell was brought to atmospheric pressure and temperature before the final readings were taken.

11 Other tests were made with the pump shut off from the condenser to determine the air leakage in the pump itself and also with the suction piping blanked off at the condenser to determine the pipe leakage.

12 In a number of cases these readings showed excessive leakage in the condenser shell which was remedied before the final readings were taken. A second apparatus has been provided for Waterside No. 1 and both stations at the present time are so arranged that in a few hours the air leakage of all the air pumps as well as the condensers may be tested. This is a great help in operating and serves a useful purpose in enabling attention to be given where there was no indication formerly of leakage.

13 A great many air tests have been made on all of the units in the two stations. Usually readings were taken during the time necessary to fill the bell (from 1 to 20 minutes) of the air temperature in the bell, the air suction temperature, air discharge temperature, vacuum at the top of the condenser, vacuum in the air suction line to the pump, hot-well temperature, circulating water inlet and outlet temperature, amount of cylinder cooling water with its inlet and outlet temperatures, revolutions per minute of the pump, power delivered by the prime mover and sometimes indicator cards on both air and steam cylinders. Table 1 shows a collection of some of the more interesting of these tests.

14 In the experimental apparatus used to obtain the data for the paper on The Transmission of Heat, it was impossible to detect any difference of temperature due to the segregation of air in the various parts of the single tube condenser. On the large condensers on which these air tests were made such

TABLE 1 RESULTS OF TESTS ON DRY AIR PUMPS FOR SURFACE CONDENSERS

Number of Tests Made	Date of Tests	Load on Turbine, Kw.	Vacuum at Top of Condenser Corrected to 30 In. Barometer	Vacuum of Pump Suction Corrected to 30 In. Barometer	Barometer	Hot Well	Circulating Water Inlet	Circulating Water Outlet	Air Suction	Air Discharge	Air in Bell	R.P.M. of Pump	Air discharged per Minute measured in Bell	Air discharged per Minute at Temperature of Air Discharge	I.H.P. Steam Cylinder	I.H.P. Air Cylinder	Mechanical Efficiency of Pump, Per Cent	Water Used to Cool Air Cylinder		
			Inches of Mercury			Temperatures, Deg. Fahr.							Cubic Feet					Temperature Inlet, Deg. Fahr.	Temperature Outlet, Deg. Fahr.	Cubic Feet used per Minute
3	2/14/11	6,500	28.49	28.61	30.09	54.7	35.7	48.8	63.0	144.7	94.3	70.0	12.94	17.15	26.92	14.80	57.1	....	46.3	....
5	2/14/11	8,200	28.43	28.61	30.07	55.0	33.0	49.8	68.0	141.7	93.3	61.4	8.59	11.08	22.85	10.36	45.7	....	52.6	....
5	2/14/11	10,400	28.16	28.40	30.04	63.0	33.0	54.2	81.2	139.4	93.1	60.4	5.82	7.40	21.52	10.24	47.6	....	53.7	....
5	2/18/11	6,266	28.50	28.60	29.92	53.0	33.6	47.9	62.8	138.2	96.5	67.2	12.65	15.65	22.51	11 94	53.0	38.56	45.0	....
5	2/18/11	8,133	28.46	28.60	29.92	57.8	33.4	51.1	73.4	136.2	96.0	66.4	6.86	8.50	21.80	9.35	42.9	38 00	45.6	2.21
5	2/18/11	10.132	28.11	28.34	29.92	68.0	33.0	54.2	84 9	137 0	95.5	65.4	5.75	7.12	20.08	10.85	58.9	38.00	46.0	2.19
3	2/18/11	10,000	28.08	28.32	29.90	68.3	33.3	54.7	85.2	144.8	97.3	74.0	5.93	7.83	23.08	12.30	53.3	37.20	43.0	3.91
2	2/18/11	10,000	27.95	28.27	29.90	69.0	34.0	56.0	86.0	144.5	96.3	72.5	5.89	7.78	23.89	11.75	49.2	38.30	49.8	....
2	2/18/11	10,000	27.90	28.23	29.90	69.5	34.5	56 0	87.7	147.3	96.0	72.0	5.27	7.07	23.32	10.58	45.3	40.00	67.8	0.49
*3	2/28/11	No load	....	29.30	30.20	....	....	....	....	99.3	98.6	75.6	1.31	....	25.32	13.99	55.3	37.70	43.0	3.14
4	3/14/11	6,075	....	29.00	30.39	....	....	....	55.0	114.0	83.5	77.0	8.28	....	....	....	....	....	....	....
6	3/14/11	9,252	....	28.99	30.35	....	....	....	57.0	111.2	88.2	75.5	6.94	....	....	....	....	....	....	....
4	3/14/11	11,613	....	28.93	30.31	....	....	....	69.8	118.9	85.5	75.5	6.33	....	....	....	....	....	....	....
*2	3/15/11	No load	....	29.10	29.98	....	....	....	....	85.7	96.5	76.5	0.63	....	....	....	....	....	....	....
	1/26/12	9,100	27.67	28.74	29.89	....	33	60	91.8	105.0	87.2	....	29.25	31.25	....	....	....	....	....	....
	1/26/12	5,970	28.15	28.65	30.03	....	33	51	91.0	100.5	88.0	....	29.60	30.95	....	....	....	....	....	....
6	4/24/11	3,022	....	28.05	30.25	....	....	....	59.1	184	95	81	39 5	....	....	....	....	....	....	....
4	4/24/11	5,204	....	28.00	30.18	....	....	....	60.6	198	98	80	38.5	....	....	....	....	....	....	....
6	4/20/11	8,258	....	28.10	29 78	....	....	....	59.0	164	93	85	36.8	....	....	....	....	....	....	....

*Valve in suction line closed.

a difference of pressure and temperature does exist, the differences of pressure being quite strongly marked. The curves for a number of these condensers, showing the increase in drop with the increase in amount of steam consumption, are illustrated in Fig. 3.

15 A number of experiments were made to find if this drop

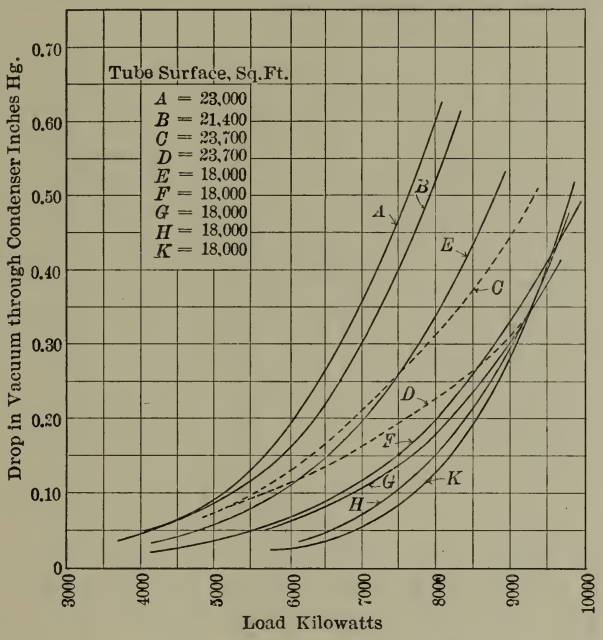

FIG. 3 CURVES SHOWING RELATION BETWEEN LOAD AND DROP THROUGH CONDENSER

was continuous through the tube surface or whether there were restrictions concentrating the drop in one or more places, and a number of experiments were made to reduce the amount of drop by taking out tubes thus opening channels into the body of the tube surface. The results here were apparently negative.

16 To ascertain the effect of known quantities of air on the vacuum suitable orifices of various sizes were installed at the top of the condenser so that the introduced air might mix with the incoming steam. These orifices were the ordinary circular orifices through thin plates and the amounts of air entering the condenser were checked up by the air delivered into the

air bell from the air pump. These results are shown in Fig. 4.

17 The tests in each case were run at constant load which was considerably below the capacity of the condenser.

18 Four sets of tests were run to determine the effect of load on air leakage, two on machines which were in bad condition and two on comparatively tight machines. The results are shown in Fig. 5. It should be noted that in every case the air leakage is less at the higher loads.

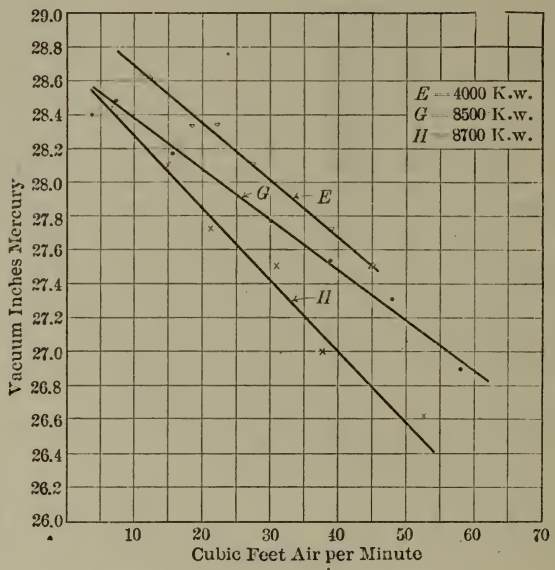

Fig. 4 Curves showing Effect of Air Leakage at Constant Load

19 In the paper on The Transmission of Heat it was inferred from Smith's curves that the heat transfer varied as the fifth power of the ratio $\frac{p_s}{p_t}$. Since that time many tests have been run from which sufficient data to calculate this relation were obtained, and the results are shown in Fig. 6. None of the tests approach the capacity of the tube surface but the results are consistent.

CONCLUSIONS

20 Where open heaters are used very little air is carried to the prime mover with the steam. The volume of this air at at-

mospheric pressure and temperature will probably not exceed 1 per cent of the volume of the feedwater. As the whole system is under pressure from the pump nearly to the final stage of the turbine no air leakage can take place up to that point.

21 The air discharged by the dry air pump at atmospheric pressure and temperature from units between 5000 kw. and 20,000 kw. in size varies from 1 cu. ft. per min. when the units are in the

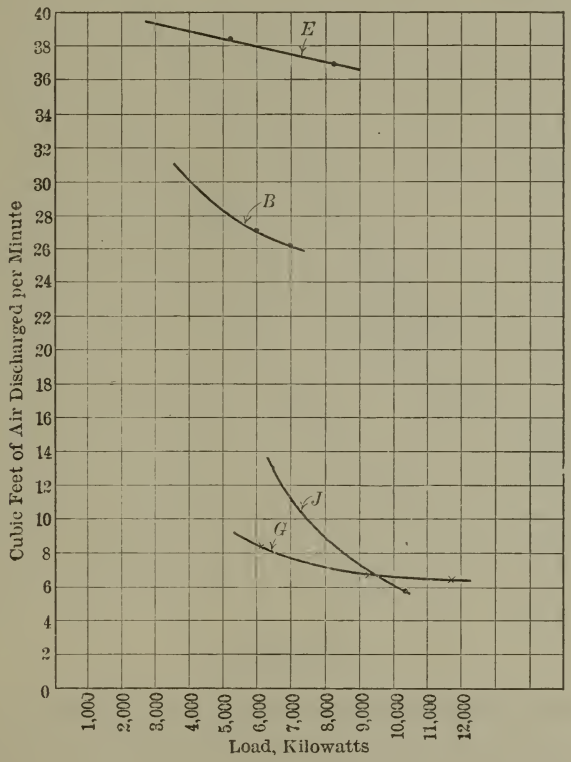

FIG. 5 CURVES SHOWING EFFECT OF LOAD UPON AIR LEAKAGE

best condition, to 15 or 20 cu. ft. where ordinary leakage is present, and to 30 or perhaps 40 or 50 cu. ft. where the units are in very bad condition.

22 Most of this leakage comes into the condenser and exhaust passages through minute leaks in the cast-iron shells, gaskets, and expansion joints. It is exceedingly difficult to detect these leaks with a candle flame but most of them may be detected by

filling the condenser with warm water under a slight head. That portion of the leakage occuring in the dry air pump and piping may be detected by the shut-off test on the pump. The volume of this leakage is larger than generally supposed and is much larger when the pump is warm.

23 The drop in vacuum through condensers of standard de-

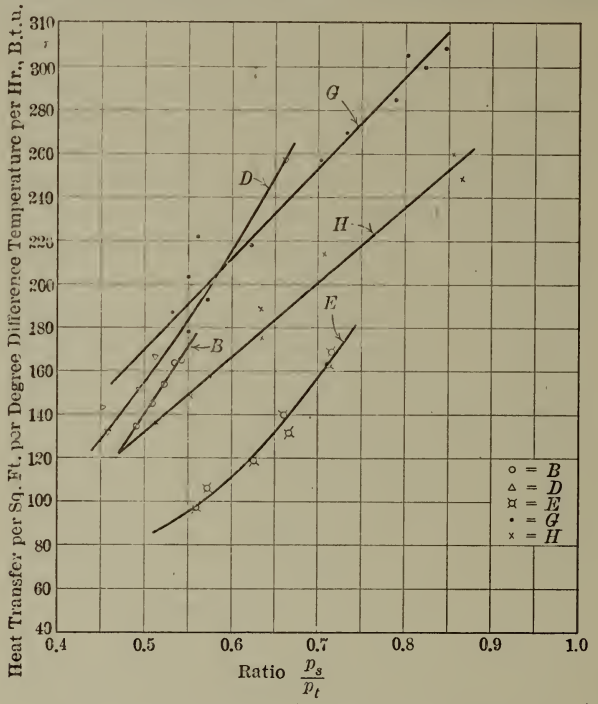

FIG. 6 CURVES SHOWING RELATION BETWEEN RATIO $\frac{p_s}{p_t}$ AND HEAT TRANSFER

sign is larger than it should be and a design which freely admits the steam to the tube surface should give better results. The opening of channels into the surface will not always improve conditions. The results indicate that wide shallow condensers should give better results although in at least two instances deep narrow condensers gave good results.

24 The falling off of vacuum with increased air leakage is considerable and the work necessary to maintain tight condensers more than pays for itself.

25 Air leakage decreases with an increase of load. This fact

in some measure corrects the decrease of vacuum at the increased loads.

26 The exponent of the ratio $\frac{p_s}{p_t}$ on page 1161 of Volume 32 of the Transactions of the Society may be changed from 5 to 2 pending further experiments to establish a better value.

27 During these experiments in which four different kinds of dry air pumps were investigated it became evident that the volumetric efficiency of a pump working between ¼ lb. and 15 lb. absolute is very poor indeed. The difficulties of keeping valves, pistons, glands and packing tight against such a tenuous fluid as air at vacuum pressures are of much greater moment than is usually believed. It is hoped that some of the types of kinetic air pumps now being placed on the market will reduce these difficulties to a practical amount.

28 Considerable difficulty was experienced in securing vacuum temperatures during these experiments and most of the work has been repeated a number of times where such temperatures are essential. It has been impossible up to the present time to secure readings which were entirely consistent and the work along this line will be carried on until the cause of the discrepancy is ascertained and correct figures obtained.

29 The author wishes to acknowledge the efficient aid rendered him in conducting these experiments by his assistants, Messrs. E. B. Rickets, J. H. Lawrence and J. S. Kerins, and by Mr. P. E. Reynolds, engineer of tests.

DISCUSSION

H. A. Everett.[1] Air has such a serious retarding effect upon condensation that any work which permits of a closer analysis of its effect is of prime importance in condenser design, and we are greatly indebted to Mr. Orrok for his work in this direction.

In 1908 D. B. Morison[2] as the result of experimental observation, said that " when the entire feedwater passes through an air pump the air in the pump discharge amounts to from 1½ to 2½ per cent by volume of the water, and that fresh water carried in tanks for use as auxiliary feed contains from 2 to 3½ per

[1] Assist. Prof. Naval Architecture, Mass. Inst. Tech., Boston.

[2] Trans. Inst. Naval Architects, vol. 50, p. 150.

cent of air by volume." These are in accordance with the values given by Mr. Orrok and are interesting as confirmatory data.

With reference to air entering by leakage there have been to date no quantitative experiments published, and those given by Mr. Orrok are of tremendous value. From them it appears doubtful if, in any case of normal leakage, the air present would be sufficient to affect appreciably the temperature of the steam at *entrance* to the condenser, though of course during the passage *through* the condenser the air richness is in ever increasing proportion as the steam is condensed out, and its effect upon condensation becomes of vital importance. The deterrent action of air on heat transference is apparently a dual one, (*a*), the insulating effect upon the tubes and (*b*), the reduction in the temperature of the steam by the reduction of the partial pressure due to the steam as the steam is condensed out and the air richness increases.

From the experiments by J. E. Smith[1] Mr. Orrok concluded in 1910 that the heat transference is proportional to the ratio of the partial pressure of the steam to the total pressure raised to the fifth power $\left(\frac{P_s}{P_t}\right)^5$ and now suggests the second power as being desirable. It should be borne in mind that the experiments of Mr. Smith were made in a condenser with no air pump, and therefore the flow of steam by the tubes was very slight. This would have the effect of permitting the zone of air around each tube to be much larger than in actual condensers where the velocity of the steam by the tubes is great, and therefore its effect would be correspondingly greater, for air is a good thermal insulator.

While this would account for the rejection of the fifth power, is Mr. Orrok warranted in suggesting two as the exponent for the ratio $\frac{P_s}{P_t}$? As from his plot of Fig. 6 the exponent is evidently considerably nearer unity than two and it is much to be desired that a description be given of the manner in which the data for the curves of Fig. 6 were taken, as knowledge of this would be of great assistance in determining the reliance to be placed upon a factor of such prime importance in his equation.

Past experience in the determination of temperatures inside of

[1] Victorian Institution of Engineers, December 1905.

condensers during operation has been most unsatisfactory, and data of this sort will be very welcome to those engaged in condenser design. In view of the apparent unreliability of the factor $\frac{P_a}{P_t}$ for determining the effect of air, would it not be desirable to drop the term from the original equation and to obtain the reduction in heat transference due to air leakage in another way? If temperatures might be predicted at different intervals down the tubing, it would be comparatively easy to estimate the actual heat transference value of surfaces located at these points from which a total or mean value could be readily derived.

A. G. Christie. Mr. Orrok is to be congratulated on his initiative in undertaking so complete a series of tests and obtaining results, as embodied in his paper, which give data and information of great value to designing and operating engineers of power plants using surface condensers. His discussion not only throws new light on some vaguely understood conditions, but also offers a solution to many questions in plant operation which have puzzled engineers. The simplicity and reliability of the methods and apparatus employed by Mr. Orrok deserve commendation. The tests have proved the effectiveness and usefulness of open heaters as air removers from boiler feedwater. This decrease in air carried to boilers not only improves the vacuum but also reduces boiler corrosion.

It is stated that most of the air comes into the condensing system through leakage in the cast-iron shells, gaskets and expansion joints. A number of years ago it was noted in a barometric condenser installation that air leakage was apparently taking place under high vacuum through the cast-iron pipes themselves as the vacuum could be improved by painting the pipes with asphaltum. The piping was well made and apparently free from blow-holes. The theory was advanced that cast iron was more or less porous to air when subjected to high vacuum. In order to test this theory some cast-iron exhaust pipes of good quality were secured from a leading condenser manufacturer and tests conducted under the writer's direction at the University of Wisconsin. It was first attempted to measure the air leakage by a method similar to Mr. Orrok's, but this was not found sensitive enough under the circumstances. Comparative results were obtained by producing a high vacuum in the piping by means of a

Hubbard pump, closing off all connections and noting the rate of decrease of vacuum. Knowing the volume of the pipes, the rate of air leakage could be computed.

It was also decided to test the effectiveness of several commercial paints or coatings for vacuum work and to note their performance under heat. A study of the results of these tests led to the following conclusions: The actual leakage of air into the pipes was very small and would not amount to any significant quantity as compared with the other volumes of air that entered through other sources. It is quite probable that practically all this leakage was through the valves, pipe connections and gaskets under the end heads. The rate of leakage was practically constant whether the pipes were bare or painted. It might therefore be said that good cast-iron pipe with a coat of suitable paint is proof against any considerable leakage of air.

Considerable trouble was experienced in preventing air leaks through the gaskets under the end heads, especially after the pipe had been heated up. Hence the bolts and gaskets at the joints should receive particular attention when air leaks are suspected. The use of cloth inserted gaskets should be avoided.

The tests were not conclusive with regard to the effectiveness of the different paints in reducing vacuum losses. Paints with asphaltum base were very effective in preventing leaks at low temperatures, but softened and tended to drip at temperatures from 210 to 225 deg. fahr., which would occur should the vacuum fail. One paint, said to be a linseed oil preparation with a peroxide of iron base, formed a close impervious film on the piping and showed no deterioration from the temperature changes within the range of the tests.

It would therefore seem that if the cast-iron portions of the condensing equipment were covered with a suitable paint, the only remaining sources of trouble from air leakage would be the bolts and gaskets at the joints.

Referring to Par. 28, the writer has frequently been puzzled to account for inconsistencies in vacuum temperatures and will welcome any new information on this subject. In one test several calibrated thermometers were tried in a well in a certain part of a jet condenser apparently out of the path of the steam, yet each showed a higher temperature than that corresponding to the vacuum shown by an adjacent mercury column. On another test the exact opposite occurred as the temperatures observed were

much too low to be accounted for by partial pressures of the air present.

In regard to the drop in vacuum through condensers, it is of interest to note European practice in condenser construction. Some British manufacturers maintain that for high efficiency of the condenser, the steam path must be so designed as to obtain high velocity through all the tubes with small loss of head. This requires a straight path through the condenser with no baffle plates and also requires a condenser with decreasing cross-section from the steam inlet. Such construction avoids dead corners for the collection of air, and at the same time, the condensed steam is swept off the tubes as fast as it forms by the high velocity. Many continental manufacturers have adopted the practice of leaving open channels through the nests of tubes to provide ready access of the steam to all cooling surfaces. As a rule baffle plates are also employed to direct the condensation to the side of the shell instead of allowing it to drip over the tubes below.

The theory advanced by several writers that seems to account in the best manner for the fall in vacuum with increased quantities of air is as follows: As steam condenses on the tube surfaces, the air is left behind and the medium surrounding the tube then contains proportionately much greater quantities of air than the steam entering the condenser. The pressure of this air lowers the partial pressure of the steam and consequently its temperature. Hence there is a less temperature difference between the tubes and the steam and therefore the rate of condensation is lowered. This theory emphasizes the necessity of providing in the design for a rapid sweeping away of the air to the air pump as the steam condenses. The high velocities maintained in some British condensers assist materially in securing this result.

William Weir.[1] As the writer's experience is chiefly associated with marine practice it differs from Mr. Orrok's on many points, but the differences may be of interest particularly as the limitations of weight and space on board ship entail the utmost care in the design of a condensing installation.

The question of air to be handled by the air pump naturally divides itself into two considerations, the amount of air entering the system through the medium of the feedwater and the amount of air entering the system as direct leakage. Apart from thermal

[1] G. & J. Weir, Ltd., Cathcart, Glasgow, N. B.

considerations, the Weir direct-contact feedheater was introduced to prevent boiler corrosion due principally to the air and other gases contained in the feedwater. This type of heater is situated on the suction side of the feed pumps, subjecting the feedwater to a reduction of pressure and an increase of temperature, which results in the water boiling and giving off the air and gas practically in their entirety. In the absence of such apparatus the feedwater will enter the boiler containing air in proportion to the feed temperature at the feed pump suction.

As regards the amount of air in feedwater at different temperatures, the writer cannot agree with Mr. J. A. Smith's curve of emissivity of gases, as in his experiments emissivity was almost entirely controlled by convection due to the stagnant condition of the experimental medium. Feedwater in any feed system is practically never still, and the actual curve of emissivity is of a totally different nature from that shown by Mr. Smith, and broadly speaking shows greater emissivity at the commencement of the heating range than at the later stages.

In turbine installations other controlling factors generally prevent the adoption of direct-contact feedwater heaters, and accordingly the feedheating is generally done on the pressure side, which does not affect to any great extent the amount of air entering with the feedwater. In such installations a large proportion of the total air enters with the feedwater. It is astonishing, however, to note that Mr. Orrok considers 15 to 20 cu. ft. of free air per minute a reasonable total air leakage on, say a 12,000-kw. installation. This seems abnormally high, and quite incompatible with good workmanship, or if associated with turbine packing glands, with good design.

The first essential in any plant designed for high vacuum is to take every possible measure to keep the air out, and the actual air leakage into such a system may be reduced in every case to a comparatively trivial amount. In marine work the condenser shells are of steel plate, and in every case the entire low-pressure system is carefully water tested after erection and before starting up, and even in land installations wherein the condensers are sometimes of cast iron, it is not difficult to maintain practically complete air-tightness, particularly in a land installation where the eduction pipe inlet to the condenser is water sealed, and where the only other pipe connections are on a simple line of air pump suction pipes. It will be obvious that

in a marine installation the difficulty is much greater owing to the extensive and complicated pipe system under the vacuum.

Mr. Orrok's experience differs also as regards air leakage at light loads, as light loads are always associated with less air leakage. It may be of interest to give an actual example of this phenomenon. Take a torpedo boat destroyer of 20,000 s.h.p. On this vessel there are two main condensers of 6000 sq. ft. each, i.e. the total cooling surface is 12,000 sq. ft.; there are two main air pumps of the Weir dual type each comprising two barrels 22 in. in diameter by 15 in. stroke. At full power with a sea temperature of 55 deg. the vacuum is 28.2 in. with both air pumps running at 50 double strokes per minute. At cruising power say 2000 s.h.p., with one air pump running at 30 double strokes on both condensers the vacuum is 28.9 in., in other words, the air to be handled by the air pump in a tight system is largely controlled by the actual amount of feedwater, and correspondingly the air brought over with it.

In connection with this example it may be of interest for American central station engineers to consider the relative proportions as regards cooling surface of these "uniflux" condensers as compared with the land practice obtaining in their country.

With regard to Mr. Orrok's remarks on the loss of pressure between the exhaust pipe and condenser base, it does not seem to be a matter of shallow as compared with deep condensers, but rather the adherence to the correct principles of design as regards permissible steam velocities, contour of shell, and pitch of tubes.

With reference to the effect of air on the actual heat transmission in surface condensers, it is apparent that this effect is different on practically every row of tubes in a surface condenser, and accordingly such experiments as Mr. J. A. Smith's are of little practical value as the element of steam flow or velocity was entirely absent.

Mr. Orrok's paper tends to confirm the writer's experience with dry air pumps of the type he has dealt with, and it has always been a matter of wonder that such types should have been adopted on any scale. All such pumps are greatly handicapped as compared with the ordinary three-valve air pump fitted with cold injection supply, and in this connection it may be of interest to refer to a paper read recently by the writer before the Institu-

tion of Engineers and Shipbuilders in Scotland,[1] wherein are given a number of particulars concerning marine practice in surface condensing installations.

J. J. Brown. In connection with the author's figures as to the amount of air entering through the prime mover, the following data, showing the amount of air entering through turbine seals, may be of interest. They were obtained by the Wheeler Condenser & Engineering Company with their portable testing gasometer, similar in design to Mr. Orrok's, and may serve to emphasize the importance of maintaining proper pressure on the steam seals.

In the test in question, which was made on a jet condenser serving a 2000-kw. turbine, at no load and with no steam on the turbine seals, a total of 86 cu. ft. of free air per minute entered the condenser, a corrected vacuum of 26.6 in. being maintained under this condition. With a normal amount of steam on the seals, and a mercury manometer registering a slight pressure, 64 cu. ft. per min. entered the condenser at no load, and the vacuum was raised to 28.5 in. On a similar unit of like size, approximately the same figures were obtained, as follows: with a load of 1650 kw. on the turbine, and no steam on the seals, 91 cu. ft. of free air per minute entered the condenser, the vacuum dropping to 25.9 in., corrected. With normal steam pressure on the seals, the amount of air was reduced to 56 cu. ft. and the vacuum increased to 28.1 in. A further increase of pressure on the seals had no appreciable effect.

It is interesting to note that the abnormal quantity of air discovered by these tests enabled the detection of leaks. Upon remedying them, the condensing plant operated well above the guarantee.

Mr. Orrok's chemical determination of the amount of air in feedwater suggests some rather unusual figures recently obtained while testing a unit. As is known, the normal quantity of oxygen in free air amounts to about 20.7 per cent. Analysis of the air extracted from a condenser revealed the fact that this amount of oxygen had been increased to an even 25 per cent, the relative quantities of CO and CO_2 remaining the same. These figures correspond to the 20 to 30 per cent of oxygen obtained by Mr. J. A. Smith,[2] in analyzing the non-condensable vapors from a condenser.

[1] November 1912.

[2] London Engineering, Oct. 7, 1904

In Fig. 3 are given curves showing the relation between load and the drop in vacuum through the condenser. Were these tests made at approximately the same vacuum, and did this vacuum remain practically constant throughout the range of load represented by each of the curves? It is well known that decreasing the pressure increases the volume; in accordance with the laws for flow of gases through orifices, by which the loss of pressure is stated as varying as the square of the velocity and inversely as the density, a loss of pressure twice as great would be expected if the velocity were doubled by doubling the volume, with the weight of steam remaining constant.

In connection with this subject of drop in pressure through condensers, experience has shown that a proper arrangement of tube surface will reduce this loss to a fraction of an inch. The most important consideration is that the incoming steam meet the maximum initial exposure of tube surface, and rectangular shells have been developed with this point in view. This is probably the nearest commercial approach to the wide shallow condensers which Mr. Orrok states should give the best results and with them during the winter season vacuums of 29.3 to 29.5 at 30 in. barometer have been maintained. The drop in pressure at the enormous volumes reached with these vacuums would be considerably larger than at 28 in., so it is evident that the loss is not excessive.

Another point to be remembered is that the volume of air increases tremendously at high vacuums. The addition of air to the total volume hence operates to cause a greater drop in pressure. While the presence of air in a condenser is detrimental under any condition, its effect is most clearly demonstrated at these high vacuums.

From the standpoint of the condenser manufacturer, it is of great value to have such a paper as Mr. Orrok's brought to the attention of engineers, and it is to be hoped that the time will come when every power station will be equipped with a gasometer or a similar device for testing for air leaks. As it is now, the condenser manufacturer must design his air pumps to handle not only the air which he knows by experience will enter with the steam or by infiltration; in addition, he must assume the responsibility for turbine seals, for careless pipe joints, and for slipshod maintenance. That is, he must proportion his air pumps to handle far more air than should be necessary, and the net result is a considerable over-all loss. It is no

exaggeration to say that probably over 75 per cent of the vacuum troubles in condensers today may be traced directly to an abnormal air leakage. It is to be hoped that further investigations similar to Mr. Orrok's will be made, in an attempt to determine the air displacement necessary, say per 1000 lb. of steam under standard conditions. The company with which the writer is connected has been working on this subject for the past two years, but if such a figure could be obtained by independent investigators it would define sharply the responsibility of the condenser manufacturer, and it would result in more careful attendance of plant and in higher efficiency, for, as Mr. Orrok points out, "the work necessary to maintain tight condensers more than pays for itself."

G. J. Foran. The entire membership of the Society is indebted to Mr. Orrok for the data presented upon the amount and variation of air leaks found in large operating units and the effect upon the results obtained.

As I understand it, Mr. Orrok presented the paper without comment on much of the data, with the idea of creating an interest and starting discussion upon the various lines. The brevity of the presentation and the absence of any discussion or demonstration of the data have already caused some misunderstanding and certain discussions which Mr. Orrok would probably not endorse.

The entire field of the effect of air upon surface or jet condensation has been the subject of special study by me for many years, but I shall confine myself only to features of unusual interest or the discussion of such data as might lead to erroneous deductions.

Referring first to the percentage of air by volume at various temperatures, as given in Par. 6, the variation in percentage corresponds quite closely to my own observations and study, and in ratio of variation, although not in exact quantity, to Winkler's tables as presented in Castell-Evan's Physico-Chemical Tables, 1911 edition. Two tables are given showing the absorption of air by water at different temperatures; the first table assuming that the pressure of the *air* outside of the liquid is constant at 760 millimeters, the second table assuming that the total pressure outside of the liquid remains constant at 760 millimeters, the pressure due to the air being the difference between the total pressure (760 millimeters) and the pressure of the aqueous vapor

due to the temperature. The tables were calculated from the absorption power of water for oxygen and nitrogen in the necessary ratio for the formation of air; the volume of air so stated is always stated in *dry* air, not saturated, a fact not definitely stated and usually overlooked.

Heretofore it has been customary in condenser work to base all calculations for the air contained in the water upon the ratio of total pressures and temperatures. It is apparent that this should be modified upon the basis of the second table referred to, and that the quantity of air present, outside of temperature considerations depends solely upon that proportion of the outside pressure due to the *air* alone.

It is well known that if a given space is saturated with aqueous vapor at a given temperature, the corresponding pressure and weight of aqueous vapor is definitely known and may be found in saturated steam tables, and this pressure and weight is absolutely independent of the presence of air or other gases. These other gases will influence the total pressure in the space, and they themselves will have a pressure and weight proportional to their portion of the total pressure; but the pressure and weight of the aqueous vapor remain the same as if there were no other gases present.

In Fig. 7 are given curves, A and B, plotted from Winkler's tables with the percentage of air laid out as ordinates and the temperatures as abscissae. A third curve, C, shows the difference between the two and curve D is plotted with inches of air pressure as abscissae. Comparing the percentage of air, as given by Mr. Orrok, in water at 187 deg. and 52 deg., the ratios agree within 4 per cent with those shown on the curve for Winkler's second table, (air—β_1). The percentage present in the hot-well water, based upon an assumed pressure in the steam space above the hot well taken by comparisons in Table 1 also corresponds very closely, that is, assuming a vacuum corresponding to a temperature 30 deg. above the hot-well temperature.

It may be claimed that the water tested was not under the absolute conditions required for the application of Winkler's tables. Strictly speaking this is true, but I believe the deviations do not seriously affect the consideration, and under any interpretation they do not approximate the ratio shown in Winkler's first table. There is not space here to discuss this question in detail, but if careful tests were made in practice, with the idea of obtaining the method of variation, I feel certain they would be found to

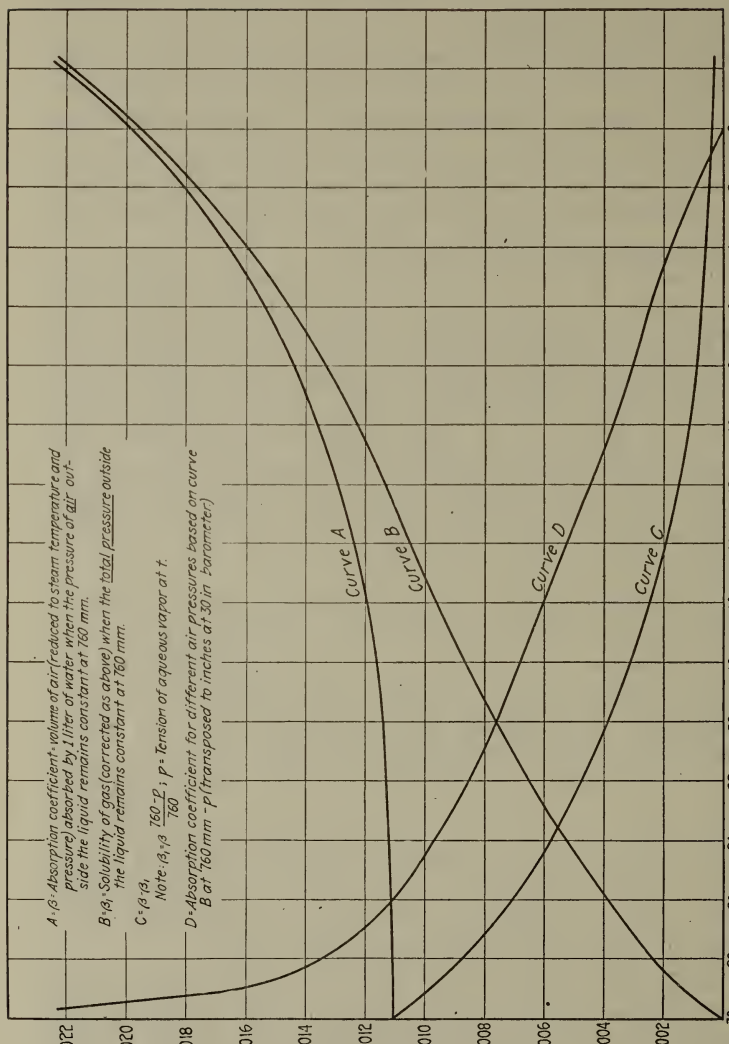

FIG. 7 CURVE SHOWING SOLUBILITY OF GASES IN WATER AT ATMOSPHERIC PRESSURE, CALCULATED FROM NITROGEN AND OXYGEN (WINKLER)[1]

[1] See table in Castell-Evans Physico-Chemical Tables, vol. 2, p. 993, ed. 1911.

conform so accurately that the curve mentioned for the variation in percentage of air present, although naturally not for the actual quantity, which varies widely, could be followed with reasonable safety.

It should be remembered that the absorption of air by water is a very slow process and does not compare with the speed of saturation of a space containing water at high temperature. We are apt to confuse these two as we are more used to considering the latter case.

Several have misunderstood Par. 7 and assumed that the temperature of saturation in the condenser always corresponds to the hot-well temperature. They never do, except by chance, their determination being due to different causes. This is shown by Table 1, the temperature in the steam space averaging more than 30 deg. above the hot-well temperature, and even the cooled vapors at the air suction being at a higher temperature than at the hot-well.

Regarding the gasometer, it is assumed that the water temperature and the air in the bell both have the external air temperature. These observations are of great interest in determining the air leakage in different plants, and of course this more or less determines the vacuum carried. However, an attempt to make too close a study of the air conditions in the condenser or in the vacuum pump, or the vacuum pump efficiency, from the observations as recorded in Table 1 should not be made, for the reason, as stated by Mr. Orrok, that in some cases the temperatures recorded are temperatures by convection from some other point of the apparatus, rather than of the air itself.

It is unfortunate that nothing definite is given as to the condition of the vacuum pumps and the vacuum they were able to maintain upon a closed suction. It is assumed that the various pumps are fitted with by-pass valves.

In Par. 27 the statement is made that the volumetric efficiency of the vacuum pump is very poor indeed. This statement should be qualified. The volumetric efficiency of such pumps is, on the contrary, very high, but in the cases observed apparently the vacuum pumps were set to operate at a speed that would give a displacement much greater than the required capacity. In that case it is evident that the unnecessary volumetric displacement would be filled with steam from the condenser, and therefore, when the comparison is made between the air in the bell and

the pump displacement, the volumetric efficiency of the pump appears poor.

It is evident that considerable steam was carried into these vacuum pumps in the manner stated. Take for example the results given on the ninth line: Here the steam cylinder is shown as developing 23.32 i.h.p., the vacuum cylinder 10.58 h.p., but the equivalent of heat taken away by the jacket water amounts to 25.8 h.p. This shows a large condensation of steam with the consequent surrender of its latent heat, and it is much greater than would correspond to the amount of saturation corresponding to the temperature and pressure carried at the suction.

The condition of the air entering the vacuum pump is of great importance. If dry air were compressed adiabatically from 2 in. to 30 in. absolute, the air having an initial temperature of 60 deg. fahr., its ultimate temperature would be 680 deg. In the case of a vacuum pump of this type, however, there is always entrained moisture carried in with the vapors, and the heat of compression as latent heat evaporates a certain portion of this moisture, thereby greatly reducing the sensible heat. This is noticeable in the tests reported. It is therefore entirely useless to attempt any discussion of vacuum pump efficiency without much more complete information, and also without some attempt to determine its most efficient speed of operation.

The benefit of operating upon a closed system and cutting out the air as much as possible is unquestionable; Fig. 8 shows the effect of stopping the vacuum pump of one of the Boston Elevated units. We realized from past experience that the vacuum upon the system would not suffer seriously even if the dry vacuum pump were stopped temporarily, and James D. Andrews, chief engineer of the plant, kindly consented to stop the vacuum pump on one of the units for ½ hour, making ½-minute observations of the fall in vacuum. This unit was developing 9000 to 10,000 kw. and carrying a vacuum equivalent to 1.03 in. absolute pressure when the pump was shut down. In ½ hour the vacuum had fallen only 0.32 in., or to 1.35 in. absolute. This curve is almost a straight line between the two points. Experiments at some length will be made later, discussing the air conditions under these circumstances.

The effect of air admitted into a condenser varies greatly with the point in the condenser at which it is admitted. It may also be admitted coincident with variations in the load and quantity

of steam, or the temperature, or quantity of the circulating water, all of which affect the results in a degree varying widely with the design of the condenser and the pressure lost through the condenser. The design of the condenser should restrict such losses as much as possible, but conditions of space and load will frequently determine absolutely the features of design.

THE AUTHOR. Whenever tests have been made of the kind reported in this paper, they have resulted in increased vacuum, less leakage and in showing the operator when and where to spend money for maintenance.

Vol. 2 of Castell-Evans tables is not generally known, and I

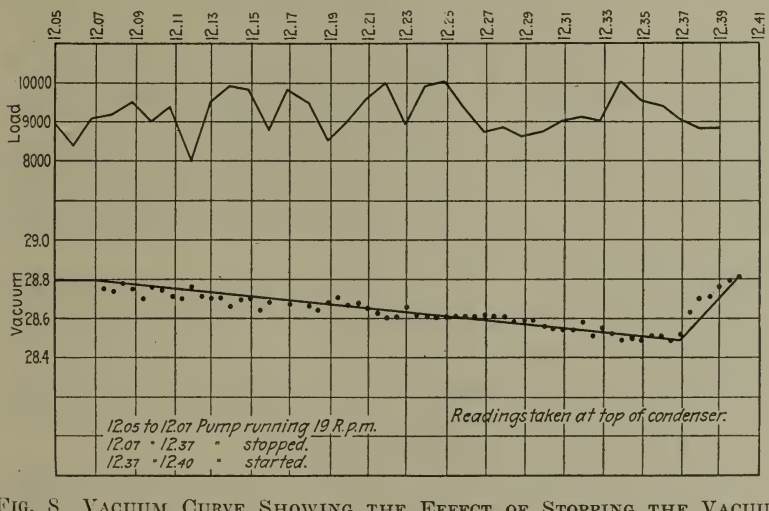

FIG. 8 VACUUM CURVE SHOWING THE EFFECT OF STOPPING THE VACUUM PUMP

thank Mr. Foran for calling attention to Winkler's table, which will be of great value to the profession.

The volumetric efficiency of air pumps depends on two things—clearance and the number of compressions. No engineer would think of compressing air to more than five or six atmospheres in one cylinder; but in the air pump with a vacuum of 29 in. we are using 30 or more compressions with correspondingly bad results in volumetric efficiency. Without doubt, all air pumps are handling more water vapor than they should, but until we can build air cylinders with less than ½ of 1 per cent clearance we must content ourselves with poor volumetric efficiency or go to two or three-stage air pumps.

Of the ten pumps referred to in the paper, six are high-speed, single, rotative valve pumps with the Weiss bypass; two are slow-speed, two-valve pumps with what corresponds to a bypass; and two are slide-valve pumps with the Weiss bypass.

Referring to Professor Everett's criticism, the curves in Fig. 6 are drawn as straight lines. The actual curves, however, must pass near the origin, and on this account the exponent (2) is offered, pending better determination. It is very simple to leave this factor out, as Professor Everett suggests, modifying the coefficient to allow for it.

Mr. Weir's light load, little air condition is strange to us where most of the air is leakage air. Even on large units the air coming over with the feed is rarely more than 1 or 1½ cu. ft. per min., and it is very rare that the total air can be brought below 3 cu. ft. per min. Practically all turbine plants are today using open or contact heaters on the suction side of the pumps.

Replying to J. J. Brown, the air in the bell is brought to atmospheric pressure by altering the counterweight and in most cases the water in the bell has so affected the air temperature that no correction need be made for this.

Referring to Table 1, the first ten lines refer to one condenser, the next four to a second, the next two to a third and the last three lines to a fourth condenser.

Referring to his third question, the curves in Fig. 3 were obtained by starting at a low load with the best vacuum obtainable and reading the drop through the condenser.

The load was progressively increased and the drops through the condenser were taken from the same instruments. During these tests, as the load was increased the vacuum decreased slightly.

No. 1364

THE PROPERTIES OF SATURATED AND SUPERHEATED AMMONIA

By William E. Mosher, Urbana, Ill.

Junior Member of the Society

The vapor of anhydrous ammonia first became of interest in the field of mechanical engineering with the advent of Carré's absorption and Linde's compression refrigerating machines. With the development of the refrigeration industry this vapor has become more and more important and an accurate knowledge of its properties is highly desirable.

2 As in the case of steam and many other vapors, the first reliable experimental knowledge of the properties of ammonia was derived from the work of Regnault. He determined experimentally the relation between the pressure and the temperature of the saturated vapor and expressed it by means of empirical formulae. He also determined the relative volumes of the superheated vapor at different pressures along an isothermal for the temperature 8.1 deg. cent., the specific heat, the theoretical density and the experimental density of the gas. The determinations made by Regnault of the specific heat of the liquid and the latent heat of vaporization were lost in the reign of the Commune in 1871; twelve of the determinations of the latter magnitude, however, have been found.

3 The tables now used most extensively in refrigeration work and included in most handbooks on refrigeration are those of Wood, based upon Regnault's experiments. Peabody's tables, which appeared at about the same time, were based upon the work of Ledoux and upon the same form of equation of state, namely that derived by Zeuner and applied to ammonia by

Presented at the Annual Meeting 1912, of The American Society of Mechanical Engineers.

Ledoux. Since these tables were published the properties of ammonia have been made the subject of considerable experimental work and various tables have been issued, although comparatively little is known of the subject even now when its importance in technical work is considered. Probably the most accurate table for the range covered is one recently calculated by Goodenough, although this was designed for classroom purposes and is not based upon refined analysis.

4 The present investigation was undertaken with the object of collecting and correlating the various scattered experimental data on the subject of the properties of ammonia. An attempt has been made to reconcile these data by means of well-known thermodynamic laws and principles so that the results may be consistent with each other; and to express the various properties by means of formulae from which tables and charts in English units may be prepared. It is realized, however, that the experimental evidence is not as complete as could be desired and that more accurate experimental work may render necessary slight revision of the values of the constants in the equations here presented.

5 *Notation.* Throughout the investigation the absolute temperature of melting ice on the fahrenheit scale has been taken as 491.64 deg., so that absolute zero is at — 459.64 deg. fahr. The value of 777.64 ft-lb. has been used for the mechanical equivalent of heat. The notation used is as follows:

J = Joule's equivalent.
A = Reciprocal of same.
t = Temperature on the fahr. or cent. scale.
T = Absolute temperature.
p = Pressure.
v = Specific volume.
γ = Specific weight.
u = Intrinsic energy.
i = Heat content at constant pressure.
s = Entropy.
r = Latent heat of vaporization.
ρ = Internal latent heat.
ψ = External latent heat = $Ap(v'' - v')$.
c = Specific heat.
c_p = Specific heat at constant pressure.
x = Quality of vapor mixture.

Subscript $_k$ indicates critical data.
$'$ indicates properties of liquid.
$''$ indicates properties of saturated vapor.

6 *Relation between Pressure and Temperature of Saturated Vapor.* To express the relation between the pressure and temperature of saturated vapors scores of formulae have been proposed, some being of more or less rational form involving empirical constants, but the greater number being purely empirical:

a Biot's equation as used by Regnault has the form

$$\log p = a - b\alpha^n + c\beta^n$$

where $n = t - k$

The same formula was used in the computation of Peabody's tables.

b Wood used Rankine's formula in the form

$$\log p = a - \frac{b}{T}$$

with $a = 6.2495$ and $b = 2196$; limits, -20 deg. fahr. to 100 deg. fahr.

c Goodenough used the Bertrand formula in the form

$$\log p = \log k - n \log \frac{T}{T - b}$$

with $\log k = 5.87395$, $n = 50$, $b = 84.3$

7 It does not seem to be possible to find constants which will make equations of the above forms apply over more than a limited range of temperature, either with steam or with other vapors. An equation for steam has, however, recently been proposed by Marks[1], which is remarkable in that it gives pressures agreeing closely with experimental values throughout the range from 32 to 706.1 deg. fahr., the latter point being the critical temperature. This equation is based on the one used by van der Waals

$$\log \frac{p_k}{p} = a\left(\frac{T_k}{T} - 1\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

where p_k and T_k denote respectively the critical pressure and critical temperature. Instead of being a constant as used by van der Waals, a varies for different substances and for different temperatures with the same substance. Professor Marks found values of a at different temperatures for steam and then found an expression for its value

[1] Trans. Am.Soc.M.E., vol. 33, 1911, p. 361.

in terms of the temperature and the critical temperature. By substituting this expression for a and the proper values for p_k and T_k in equation [1], he arrived at the following relation between the pressure and temperature of saturated steam

$$\log p = a - \frac{b}{T} - cT + eT^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

where

$a = 10.515354$
$b = 4873.71$
$c = 0.00405096$
$e = 0.000001392964$

8 In the present investigation an attempt was made to use the above method to determine the pressure-temperature relation of ammonia. Several factors rendered the use of this method undesirable, among them being the sensitiveness of the method to changes in the critical data and the uncertainty of these data for ammonia, the lack of experimental data at temperatures near the critical point, and the large discrepancies existing in the data throughout the entire range. As will be seen later, however, this equation plays an important part in the present investigation.

9 Physicists have repeatedly attempted to find a relation between the pressures of different vapors at the same temperature such that a determination of the function $p = f(t)$ for one vapor would serve to determine this function for all other vapors. Ramsey and Young proposed the law

$$R = R' + k\,(T - T')$$

where R and R' are the ratios of the temperatures of two saturated vapors at two different pressures, and T and T' are the temperatures of one of the vapors corresponding to these pressures. They show that the law holds very closely for some 22 different substances arranged in 23 different pairs. In their paper they use the T and T' in the term $k\,(T - T')$ as being indiscriminately the numerators or the denominators of the ratios R and R'; i. e. they use the two equations

$$\frac{T_b}{T_a} = \frac{T'_b}{T'_a} + k'\,(T_a - T'_a) \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

$$\frac{T_a}{T_b} = \frac{T'_a}{T'_b} + k\,(T_a - T'_a) \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

without making any mention of the change from one form to the

other and without recognizing that the two forms lead to different results.

10 This law has been corroborated by the work of Richardson and by the work of Ramsey and Travers on crypton, argon and xenon.

11 Ayrton and Perry, and Everett have each remarked upon the lack of symmetry of equation [3] and have shown that equation [4] is symmetrical. This is most clearly demonstrated by Moss who shows that [3] may be thrown into the form

$$\frac{T_b}{T_a}=\left[\frac{T_b'}{T_a'}-k'T_a'\right]+k'T_a$$

and [4] may be thrown into the form

$$\frac{T_a}{T_b}=\left[\frac{T_a'}{T_b'}-kT_a'\right]+kT_a$$

since T_a' and T_b' are the temperatures corresponding to some particular vapor pressure, these equations may be written respectively

$$\frac{T_b}{T_a}=k'T_a+c' \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [5]$$

and

$$\frac{T_a}{T_b}=kT_a+c \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [6]$$

To test the symmetry of these equations they may be written respectively

$$\frac{T_a}{T_b}=\left(-\frac{k'}{c'}\right)\frac{T_a^2}{T_b}+\frac{1}{c}$$

and

$$\frac{T_b}{T_a}=\left(-\frac{k}{c}\right)T_b+\frac{1}{c}$$

It is seen that if the left-hand member of [5] is a linear function of the denominator its reciprocal will not be a linear function of the new denominator. On the other hand if the left-hand member of [6] is a linear function of the numerator, its reciprocal will also be a linear function of the new denominator.

12 Equation [6] may be written in the simple form

$$\frac{1}{T_b}=c\,\frac{1}{T_a}+k \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [7]$$

$\frac{1}{T_b}$ and $\frac{1}{T_a}$ thus being linear functions of each other.

13 Since equation [7] may be written

$$A-\frac{B}{T_b}=A'-\frac{B'}{T_a}$$

it follows that any equation to be applicable to all vapors in the same form with only its constants changed, and at the same time to be consistent with the temperature ratio law as stated above, must satisfy the condition that

$$p=f\left(A-\frac{B}{T}\right)$$

all other constants remaining the same for all vapors. Now, it happens that the equations which may be thrown into this form, such as the Rankine short form, the Roche equation, etc., do not satisfy the experimental data for steam throughout its range with the necessary degree of accuracy. On the other hand, the Marks equation, which does satisfy these data with remarkable accuracy throughout the complete range, cannot be thrown into the required form. In view of the various considerations mentioned it has been decided in the present investigation to accept the temperature ratio law as expressed in equation [7], to use water as the standard substance, to use the Marks equation, equation [2], as representing the pressure-temperature relation for water, and to make a step-by-step solution of the pressure-temperature relation for ammonia by means of these equations.

14 The method used in applying the temperature ratio law and in determining the value of the constants c and k in equation [7], is that used by Moss, who has applied it to 17 different vapors. A description of this method and of the construction of Fig. 1 is given in Appendix No. 1.

15 After plotting the points representing the experimental data the straight line shown in Fig. 1 was drawn in a manner such as to represent all the points in the best possible manner.

16 The effectiveness of any method that enables one to throw any given data into a form such that it may be represented by a straight line lies in the fact that it discloses immediately and strikingly any departure from the general trend and reveals unerringly points that depart from this trend. It enables one to give various observations their proper weight and it affords a much safer basis of extrapolation than can be obtained from curves. The case in hand illustrates this principle. If the points are plotted on the regular pt plane it is easily seen that no smooth curve such as represents a law of nature, could, if passed through Regnault's lower temperature points and

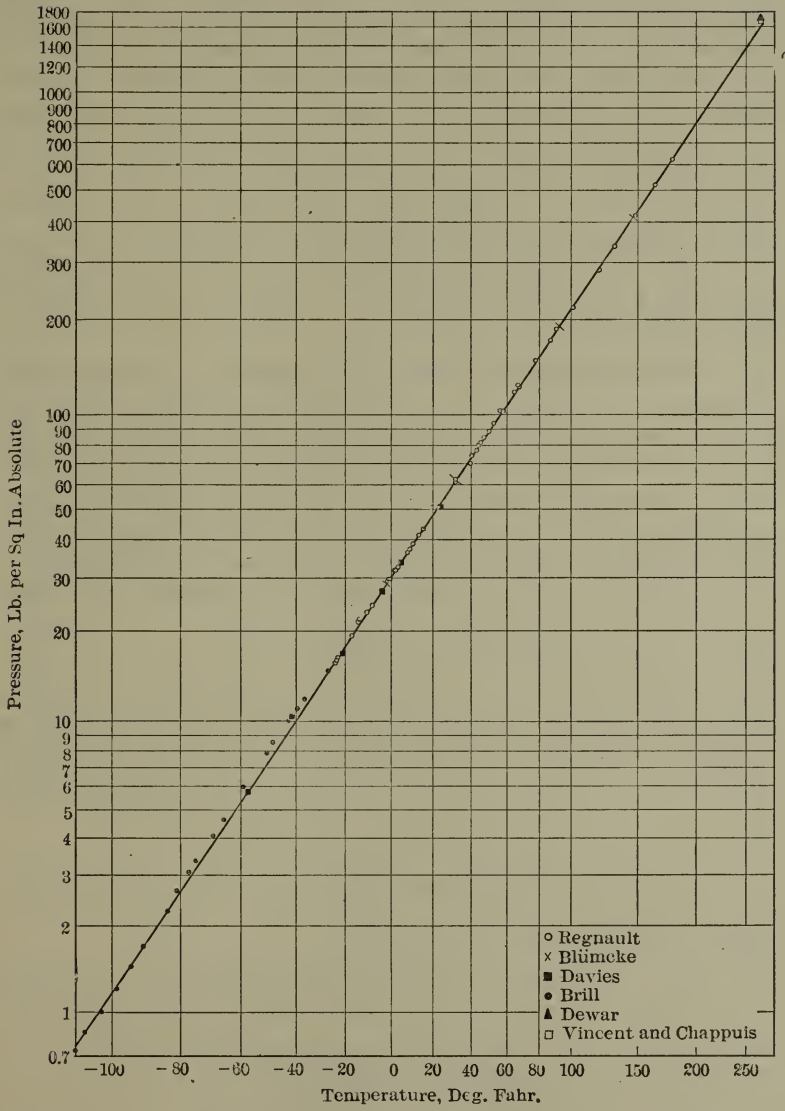

FIG. 1 CURVE SHOWING AGREEMENT OF PRESSURES FOUND BY TEMPERATURE RATIO LAW WITH THOSE FOUND BY EXPERIMENT

Brill's lower points, at the same time pass through Brill's points in the region −80 to −30 deg. fahr. Further than this the chart discloses nothing, as an infinite number of curves could be drawn, giving different weights to the different points, or all points could be given equal weight and the equation could then be determined by least squares. An inspection of Fig. 1 shows, however, that the straight line which best represents the whole range represents very accurately Regnault's lower points, Brill's highest point and all of his points from −80 to −110 deg. fahr. In the range −80 to −30 deg. fahr. all of Brill's points lie above the line, as does Davies' −41.8 deg. fahr. point. It is significant, however, that Davies' −57.64 deg. fahr. point lies exactly on the line at precisely the point where Brill's points lie farthest from it.

17 Through the range −30 to +100 deg. fahr. the line represents the experimental points very accurately; above 100 deg. fahr. Regnault's points lie slightly below the line. Now all of Regnault's points above 100 deg. fahr. belong to his third series of experiments. In all of the higher pressure points of this series Regnault used a closed manometer and extensive corrections had to be applied to the height of mercury observed, and a comparison of the experiments in the range 40 to 90 deg. fahr. where the second and third series overlap shows that invariably the points determined in the third series by the use of the closed manometer lie below those determined in the second series by the use of the open manometer. The conclusion is evident that there was probably an error in the correction applied by Regnault to his readings with the closed manometer; the fact that these points lie below the line should therefore not be taken as conclusive evidence of the incorrectness of the line

18 As to the critical data, it is seen that the points of both Dewar, and Vincent and Chappuis lie above the line. If the line be taken as correct this indicates either higher temperature, lower pressure, or both, at the critical point.

19 This is also the case with the determination of the critical data for water; as methods have been improved and greater precautions taken in the experimental work, the value found for the critical temperature has steadily risen until the latest determination, that of Holborn and Bauman, is 30 deg. higher than that found by Nadejdine 25 years earlier. It is reasonable to assume that the same errors may have been made in the ammonia determinations and that the true critical temperature of ammonia is higher than that found by either Dewar or Vincent and Chappuis. In the

present investigation the critical data of ammonia are not of prime importance and the purely arbitrary assumption has been made, on the basis of the above reasoning and for want of better data, that the higher of the critical pressure determinations, that of Dewar, is correct. The corresponding temperature as determined by the temperature ratio law has been taken as the critical temperature. The resulting values are

$$p_k = 1690 \text{ lb. per sq. in.}$$
$$t_k = 273.2 \text{ deg. fahr.}$$

20 Since the above conclusions were drawn the articles giving the determinations of Jaquerod and Scheffer have been found. If plotted in Fig. 1 the point representing Jaquerod's determination falls below the straight line and that representing Scheffer's value falls almost exactly upon it. If Scheffer's value for the critical temperature is substituted in the temperature ratio law the resulting pressure is 1638.6 lb. per sq. in. while his experimental value is 1635.7 lb. per sq. in., the difference being about 1/6 of one per cent. These results offer corroboration of the correctness of the location of the straight line in Fig. 1, although the arbitrary values chosen for the critical temperature and pressure may be too high.

21 The equation of the straight line in Fig. 1 is

$$\frac{1}{T_a} = 1.70356 \frac{1}{T_w} - 0.0002242 \quad \ldots\ldots\ldots\ldots\ldots\ldots [8]$$

In constructing tables with pressure as the argument the temperature of saturated steam at any pressure is found from steam tables. The corresponding saturation temperature of ammonia is then found by using equation [8] in the form

$$T_a = \frac{1}{\frac{1.70356}{T_w} - 0.0002242}$$

22 If temperature is taken as the argument the temperature at which steam will be saturated under the same pressure may be found by using equation [8] in the form

$$T_w = \frac{1}{\frac{0.587006}{T_a} + 0.0001316}$$

and the corresponding pressure is found from steam tables.

23 *Specific Volume of the Liquid.* There are available three sets of experimental data from which the specific volume of the liquid may be determined. Lange determined the density of liquid ammonia over the temperature range of −56 to +208 deg. fahr.; Diet-

erici, working by Young's method, obtained simultaneously the specific volumes of the liquid and of the saturated vapor over a temperature range of 32 to 222 deg. fahr. In addition to these there are the specific gravity determinations of d'Andréeff, from which the specific volume may be calculated; these experiments cover the range of 14 to 68 deg. fahr.

24 In Fig. 2 the experimental values are plotted with specific volumes as ordinates, and temperatures as abscissae. The form of equation used to express the volume of the liquid in terms of the temperature is that used by Avenarius, or

$$v' = a - b \log (t_k - t) \quad [9]$$

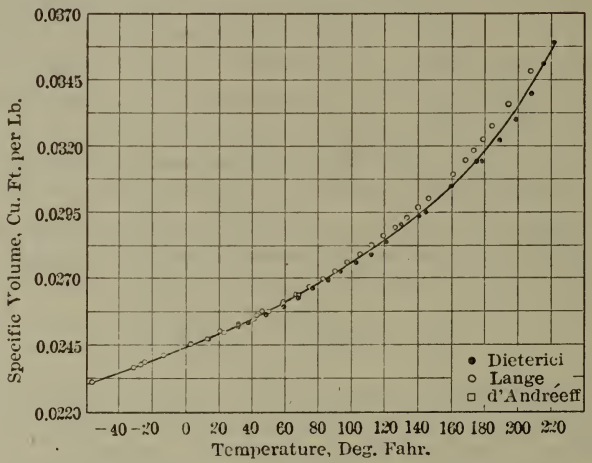

FIG. 2 CURVE SHOWING AGREEMENT OF VOLUMES OF LIQUID AS CALCULATED WITH THOSE FOUND BY EXPERIMENT

The curve shown in Fig. 2 up to 160 deg. fahr. represents this equation with the constants as follows:

$a = 0.06335$
$b = 0.016$
$t_k = 273.2$

25 In determining these constants the work of Dieterici was given the most weight, both because the methods he employed are superior to the others, and because the volumes of the vapor to be used with these values are those determined by Dieterici by the same method in the same series of experiments.

26 The part of the curve above 160 deg. fahr. represents the values found by the method discussed in Pars. 45 to 50.

27 *Latent Heat of Vaporization.* Since the experimental information regarding the latent heat of vaporization of ammonia is too meager to be used as a basis for the determination of the relation existing between it and the temperature of vaporization, this relationship will be determined from other considerations and the available data used as a check on the method.

28 Let [7] be differentiated with respect to the pressure. Then

$$\frac{-\frac{dT_a}{dp}}{T_a^2} = \frac{-c\frac{dT_w}{dp}}{T_w^2}$$

which may be written

$$\frac{T_a\frac{dp}{dT_a}}{T_w\frac{dp}{dT_w}} = \frac{1}{c}\cdot\frac{T_w}{T_a}$$

or since

$$\frac{T_w}{T_a} = c + kT_w$$

there results

$$\frac{T_a\frac{dp}{dT_a}}{T_w\frac{dp}{dT_w}} = 1+\left(\frac{k}{c}\right)T_w \quad \ldots\ldots\ldots\ldots\ldots\ldots [10]$$

Now by the Clapeyron relation connecting the latent heat and the absolute temperature of vaporization with the change of volume during vaporization

$$\left(\frac{v''-v'}{r}\right) = \frac{1}{144AT\left(\frac{dp}{dT}\right)_{sat}}$$

Substituting in equation [10]

$$\frac{\frac{(v''-v')_w}{r_w}}{\frac{(v''-v')_a}{r_a}} = 1+\frac{k}{c}T_w \quad \ldots\ldots\ldots\ldots\ldots\ldots [11]$$

or with the proper values of the constants introduced

$$\frac{\dfrac{(v''-v')_w}{r_w}}{\dfrac{(v''-v')_a}{r_a}} = 1-0.0001316T_w \quad \ldots\ldots\ldots\ldots [12]$$

29 The quantity $\dfrac{(v''-v')_w}{r_w}$ may be found from steam tables or as follows: From the Clapeyron relation

$$\frac{(v''-v')_w}{r_w} = \frac{1}{144\,AT\left[\dfrac{dp}{dT_w}\right]_{sat}}$$

By differentiating [2]

$$\left[\frac{dp}{dT_w}\right]_{sat} = p\left[\frac{11222.13}{T_w^2}+0.00000641484\ T_w-0.00932768\ T_w^2\right]$$

Substituting this value in the Clapeyron relation

$$\frac{(v''-v')_w}{r_w} = \frac{1}{p\left[\dfrac{2078.07}{T_w}-0.00172726\ T_w+0.00000118787\ T_w^2\right]} \quad [13]$$

30 By the use of equations [12] and [13] $\dfrac{(v''-v')a}{r_a}$ may be calculated and if either numerator or denominator is known the other may be found. Fortunately there are available the experimental determinations of Dieterici of the specific volumes of the liquid and saturated vapor; the former were given in Pars. 23–26 and the latter will be discussed in Pars. 41–44. From these determinations various values of $(v''-v')$ were found and values of r calculated as described above.

31 According to the generally accepted ideas concerning the critical point, at that point $r=0$ and $\dfrac{dr}{dt} = -\infty$. These facts led Thiesen to suggest as an empirical formula

$$r = c\ (t_k-t)^n$$

This form of equation has been used for water by Thiesen, Henning, and Marks and Davis, and for ammonia by Dieterici. When plotted

on logarithmic cross-section paper it is represented by a straight line, the slope being equal to n.

32 The values of r found by the above method were plotted to the corresponding values of $(t_k - t)$ on logarithmic cross-section paper and were found to lie almost exactly on a straight line, indicating that the Thiesen formula might be used to express the desired relation. The formula may be written in the form

$$\log r = \log C + n \log (t_k - t)$$

and it was found that the values of the constants giving the best agreement with the plotted points are

$$\log C = 1.856064$$

and

$$n = 0.37$$

Therefore the final equation is

$$\log r = 1.856064 + 0.37 \log (273.2 - t) \ldots\ldots\ldots\ldots [14]$$

33 The available data concerning the latent heat of vaporization are given in Table 1 and the various points are plotted in Fig. 3, the full line in this figure representing equation [14]. Of the determination of r made by Regnault, the greater number were lost, but 12 were saved and later published. The results of these experiments do not give r directly and have been variously interpreted by different writers. The table contains the interpretation of Jacobus and the three values of r as quoted from Regnault by Dieterici. Two values for r are given by von Strombeck, one being an average of 12 experiments, the other of eight. The value given by Franklin and Kraus is an average of three determinations at the normal boiling point, and the value obtained is exactly the value deduced by the same writers from the absolute boiling point and the molecular elevation. A different value was found at the normal boiling point by Estreicher and Schnerr. The values given by Denton and Jacobus, represented in Fig. 3 by crosses, were calculated from readings taken during a test of an ammonia compressor, and hence cannot be considered of much weight; because of the scarcity of scientific data, however, these points are included in the table and chart.

34 In Fig. 3 are also plotted the curves that represent various equations used in calculating the latent heat of vaporization, as given in the various existing tables. These are plotted exactly as given and it should be remembered that there is some variation in the value of the heat unit used by different writers; consequently the

various curves are not exactly comparable and are given simply to show the variation in the values of latent heat as given in the tables now in use.

TABLE 1 SUMMARY OF VARIOUS DETERMINATIONS OF THE LATENT HEAT OF VAPORIZATION

Temperature, Deg. Cent.	Cal. per Kg.	Temperature, Deg. Fahr.	B.t.u.	Authority
10.90	287.0	51.62	516.6	
15.53	285.2	59.95	513.3	
16.00	290.5	60.80	522.8	
12.94	283.8	55.29	510.8	
11.90	285.8	53.42	514.4	
10.73	288.1	51.31	518.5	Regnault
11.04	292.5	51.87	526.4	(Jacobus)
10.15	292.4	50.27	526.2	
9.52	295.0	49.14	531.0	
10.99	293.3	51.78	527.9	
12.60	291.6	54 68	524.8	
7.80	291.8	46.04	525.2	
7.80	294.2	46.04	529.5	
11.00	291.3	51.80	524.3	Regnault (Dieterici)
16.00	297.4	60.80	535.3	
19.53	296.5	67.15	533.7	Von Strombeck
17.00	296.8	62.60	534.2	
−33.64	337.0	−28.55	606.6	Franklin and Kraus
−33.40	321.3	−28.12	578.3	Estreicher and Schnerr
.....	.....	84 6	524.8	
.....	.....	82.7	525.7	
.....	.....	87.7	512.4	Denton and Jacobus
.....	.....	−10.7	569.2	
.....	.....	− 3 2	603.5	
.....	.....	+14.5	570.4	

35 The equations of the curves shown in Fig. 3 are

Ledoux ..... $r = 583.33 - 0.5499t - 0.0001173t^2$
Wood ..... $r = 555.5 - 0.613t - 0.000219t^2$
Peabody ..... $r = 540 - 0.8\ (t - 32)$
Dieterici ..... $\log r = 1.56141 + 0.5 \log (266.9 - t)$
Goodenough ..... $\log r = 1.7920 + 0.4 \log (266 - t)$

36 No comment is necessary regarding the curves for r according to Ledoux, Wood and Peabody, since their equations are not of a form such as to give correct results at high temperatures. Moreover they were derived before there were experimental determinations of the volume of the saturated vapor, from which latent heats could be calculated.

37 Dieterici used Zeuner's empirical expression for the second term of Regnault's pressure-temperature equation, and omitted the third term entirely. This approximation is quite accurate for low temperatures, but as the temperature increases the pressures thus calculated become too small, and this error increases rapidly at high temperatures. For instance, Regnault extrapolates his curve and gives for 212 deg. fahr. a pressure of 901.6 lb. per sq. in., while the

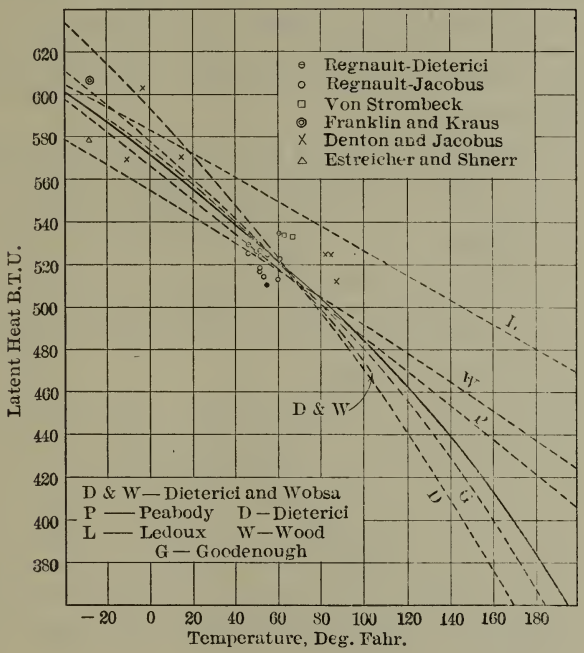

FIG. 3 CURVES SHOWING COMPARISON OF EXPERIMENTAL VALUES FOR LATENT HEAT OF VAPORIZATION WITH VALUES AS CALCULATED BY VARIOUS WRITERS

value calculated by Dieterici for this temperature is 847.1 lb. per sq. in. or 6 per cent lower. As the error is increasing, the values of $\left[\frac{dp}{dt}\right]_{sat}$ are too small and values of r calculated by means of the Clapeyron equation are too small at high temperatures. Since the relation employed between r and $(t_k - t)$ is a straight line on logarithmic paper and the values of $\frac{dp}{dt}$, consequently of r, agree at some

medium temperature, the values given by the equation at low temperatures will be too high. This is shown in Fig. 3, where it is seen that Dieterici's curve passes above even the highest of the actual experimental determinations of r at -28 deg. fahr.

38 The equation used by Goodenough fits the actual determinations and is consistent with the Clausius relation, using Dieterici's values for volumes and Goodenough's constants in Bertrand's pressure-temperature equation. This latter equation, however, applies over but a small temperature range. The variation between Goodenough's r curve and the present one is due to the different pressure-temperature relation used and the higher value assumed for the critical temperature.

39 It is believed that, due to the form of equation [14] and the accuracy with which it represents the derived values of r where known, the equation may be safely extrapolated as far as required in this investigation.

40 The comparison between the values of r given by equation [14], the corresponding values of v'' derived from the Clausius relation, and the experimental values for v'', properly belongs in the next section and is there shown.

41 *Specific Volume of Saturated Vapor.* The only available experimental determinations of the specific volume of the saturated vapor are those of Dieterici. These experiments extend up to a temperature of 222 deg. fahr., but unfortunately were not carried below a temperature of 32 deg. fahr. Owing to the form of the curve representing the volume-temperature relation, any extended extrapolation of the curve or of any empirical equation to represent this relation would be very unsafe below 32 deg. fahr., where the volume is seen to increase very rapidly.

42 As shown in the preceding section, however, equation [14] is believed to be of a form such that extrapolation can be carried to very low temperatures. Therefore in the present investigation the volumes of the saturated vapor have been calculated as follows:

43 By the use of equations [12] and [13] values of $\frac{(v''-v')_a}{r_a}$ were calculated. These values multiplied by the corresponding values of r found from equation [14] gave the values of $(v''-v')$. The addition of the proper values of v' found from equation [9] gave the values of v'', the quantity desired.

44 The agreement of the values of v'' calculated by the above

method is shown in Fig. 4, the curve in this figure representing the calculated values and the points the observed values.

45 *Specific Volume of Liquid and Saturated Vapor at High Temperatures.* The "law of the straight diameter" was first proposed by

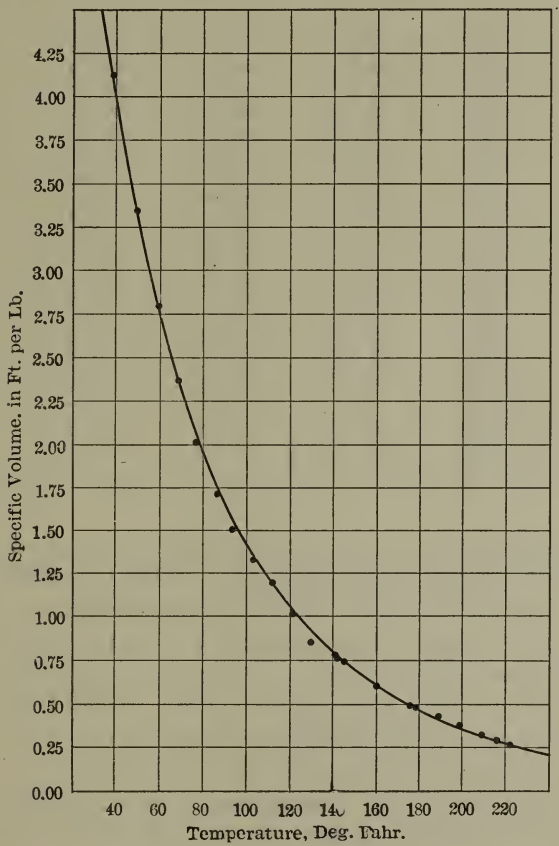

FIG. 4 CURVE SHOWING AGREEMENT OF VOLUMES OF SATURATED VAPOR AS CALCULATED WITH THOSE FOUND EXPERIMENTALLY BY DIETERICI

Cailletet and Mathias in 1886. This law, as originally given, stated that if the densities of a liquid and its saturated vapor are plotted (as abscissae) against the corresponding temperature (as ordinates) to form a dome, the mid-points of the horizontal chords of this dome will lie in a straight line nearly parallel to the axis of temperatures. References to the law may be found in an article by Davis.[1]

[1] Proc. Am. Acad. of Arts and Sciences, vol. 45, 1910, p. 305.

TABLE 2 THE LAW OF THE STRAIGHT DIAMETER FOR AMMONIA

Temperature, Deg. Fahr.	Density of Vapor	Density of Liquid	Mean Density	By Formula	Difference
−40	0.0393	42.704	21.372	21.372	±0.000
−30	0.0517	42.296	21.174	21.174	±0.000
−20	0.0671	41.883	20.975	20.974	−0.001
−10	0.0860	41.461	20.774	20.774	±0.000
0	0.1088	41.039	20.574	20.572	−0.002
+10	0.1362	40.608	20.372	20.370	−0.002
20	0.1689	40.169	20.169	20.167	−0.002
30	0.2075	39.722	19.965	19.964	−0.001
40	0.2525	39.267	19.760	19.759	−0.001
50	0.3051	38.803	19.554	19.554	±0.000
60	0.3657	38.329	19.347	19.347	±0.000
70	0.4355	37.844	19.140	19.140	±0.000
80	0.5152	37.348	18.932	18.932	±0.000
90	0.6061	36 841	18.724	18.724	±0.000
100	0.7099	36 319	18.514	18.514	±0.000
110	0 8265	35.782	18.304	18.303	−0.001
120	0.9590	35 228	18.093	18.092	−0.001
130	1.1062	34.655	17.881	17.880	−0.001
140	1.2756	34.061	17.668	17.667	−0.001
150	1.4643	33.444	17.454	17.453	−0.001
160	1.6763	32.798	17.237	17.238	+0.001

46 It was found by Young that the diameter is actually straight in the case of all but a few substances, normal pentane being an example of this class. In the case of most substances, however, the diameter can be represented accurately by a second degree equation; some substances, such as alcohols, require a third degree equation.

47 In order to find the equation of the straight diameter for ammonia the values of the densities of the liquid and of the saturated vapor were found at 10 deg. intervals. The values for the liquid were calculated by the use of equation [9], and those for the vapor by the method described in Pars. 41–44. At each of the temperatures the mean density was calculated. The results are given in Table 2 and plotted in Fig. 5, where the points for the liquid and vapor are shown as large circles and the mean points as small circles. The diameter is seen to be slightly curved, but it may be accurately represented by the second degree equation

$$\gamma = 19.3473 - 0.02067\ (t-60) - 0.0000042\ (t-60)^2 \ldots\ldots [15]$$

The table shows that this equation represents the mean densities as calculated by the other method with a maximum error of about

1/100 of one per cent over a temperature range of −40 to +160 deg. fahr.

48 The method used at ordinary temperatures for finding the change of volume on vaporization, or $(v''-v')$, may be employed at temperatures up to the critical point if it is assumed that the values

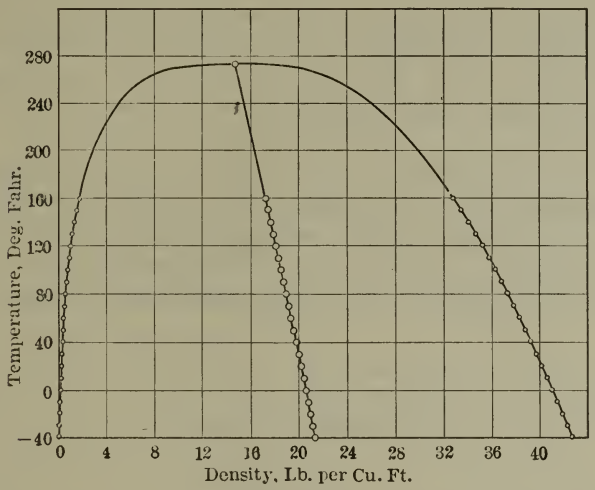

FIG. 5 CURVE SHOWING DOME OF TEMPERATURE-DENSITY PLANE AND "STRAIGHT DIAMETER" OF CAILLETET AND MATHIAS

for r found by equation [14] are correct up to that point. Equation 15] gives values for

$$\frac{\gamma''+\gamma'}{2}=\frac{1}{2}\left[\frac{1}{v''}+\frac{1}{v'}\right]$$

A simple algebraic manipulation gives

$$v''=\frac{\gamma(v''-v')+1+\sqrt{\gamma^2(v''-v')^2+1}}{2\gamma}$$

Having values for v'' and $(v''-v')$, the values for v' may be easily found. This method enables the values of the volumes of the liquid and the saturated vapor to be found at least qualitatively up to the critical point itself.

49 Above 160 deg. fahr. the values found for v' by this method were materially lower than those found by equation [9] and that

the difference increased as the temperature increased above this point. An inspection of equation [9] shows that this equation would give a value of $+\infty$ for the volume of the liquid at the critical point, whereas at this point the liquid has a finite volume. In the case of other vapors it has been shown that the form of the equation used, that of Avenarius, gives accurate values at temperatures somewhat removed from the critical point, but that it does not hold near this point. Therefore, in the present case equation [9] has been used only up to 160 deg. fahr., and the values above this temperature have been calculated by the law of the straight diameter.

50 Since the diameter is nearly parallel to the axis of temperatures a considerable error in the value of the critical temperature will cause but a small error in the resulting value of the critical density. The substitution in equation [15] of the assumed value of t_k, or 273.2 deg. fahr., gives for the critical density and volume respectively

$$\gamma_k = 14.75 \text{ lb. per cu. ft.}$$
$$v_k = 0.0678 \text{ cu. ft. per lb.}$$

51 *Specific Volume of the Superheated Vapor.* The attempt has often been made to deduce rationally an equation of state which with suitable change of constants will represent the relation $f(P, v, T) = 0$ for various fluids in all states from the gaseous condition above the critical temperature to the liquid condition. Such equations are constructed with special reference to the behavior of fluids in the neighborhood of the critical state and apply more particularly to fluids, the critical temperature of which is within the range encountered in practice. In the case of a fluid such as ammonia, however, the critical temperature of which is far above the working range, purely empirical equations of simpler form give better results throughout the small range covered in practice, and in addition lend themselves much more readily to the formation of the various heat equations. Of these equations those of Ledoux, Zeuner, Peabody and Wood were based upon various doubtful assumptions, while Wobsa's first equation gives values that do not closely agree with experimental results, and only his second equation

$$v - a = \frac{BT}{p} - \frac{c}{T^n} + \frac{b}{p} \quad \text{[16]}$$

where

$B = 49.736$
$a = 0.0075$
$C = 2450$
$n = 2$
$b = 80$

is quite satisfactory in this respect.

52 In choosing a characteristic equation several points must be considered:

a The equation must represent with fair accuracy the available reliable experimental data on the relations of p, v and T.

b The equation should be of such a form as to make the various thermodynamic relations derived from it as simple as possible.

c These derived equations must represent accurately the experimental data.

53 Wobsa's second equation fulfils the first of these conditions admirably. It is, however, somewhat defective with respect to the other requirements. The good results obtained from Goodenough's equation for superheated steam

$$v + c = \frac{BT}{p} - (1 + ap)\frac{m}{T^n} \quad \ldots\ldots\ldots\ldots\ldots\ldots [17]$$

suggested the adoption of the same form of equation for superheated ammonia in the preliminary investigation. This equation satisfies the second condition in that it gives derived relations of comparatively simple form. A trial with various sets of constants showed that it could be made to represent the volume measurements substantially with the same accuracy as Wobsa's second equation, and thereby satisfy the first condition. Having established the fact that the proposed equation is permissible, the next step was the determination of the constants. In connection with this process emphasis must be placed on the third consideration heretofore mentioned. From the characteristic equation are derived expressions for (*a*) the specific heat at constant pressure; (*b*) the heat content of the superheated and also of the saturated vapor; (*c*) the Joule-Thomson coefficient. Hence the constants must be chosen not with reference to volume alone. While the volume measurements must be satisfied,

three other derived relations must conform equally well to the experimental data in the respective fields.

54 With due consideration of all the conditions the following values were finally assumed for the constants:

$B = 0.6321$, p in lb. per sq. in.
$\log m = 12.900000$
$c = 0.100$
$n = 5$
$a = 0$

The final equation with constants inserted is therefore

$$v + 0.100 = 0.6321 \frac{T}{p} - \frac{79433 \times 10^8}{T^5} \quad \text{[18]}$$

55 It will be seen that the constant a is taken as zero. In view of the fact that the available data consist only of values along the saturation curve and in most cases but one point on each isotherm in the superheated region, no information is available regarding the shape or curvature of these isotherms on the $pv-p$ plane; moreover the derived equations demand an exceedingly small value of a. The use of straight lines for these isotherms therefore seems to be as well justified as the use of parabolas; hence a was made equal to zero.

56 A summary of the various direct determinations of the specific volume of the superheated vapor according to Perman and Davies and LeDuc and Guye is given in Table 3. In Table 4 the values given by equation [18] and by Wobsa's first and second equations are compared with these experimental determinations. It is seen that both equation [18] and Wobsa's second equation give results agreeing better with experiment at the one atmosphere points than those obtained from Wobsa's first equation. At the 32, 122 and 212 deg. fahr. points at one atmosphere pressure equation [18] and Wobsa's second equation give practically the same percentage deviation from the experimental values. At -4 deg. Wobsa's second equation gives a better agreement with the experimental value than does equation [18]; this point, however, is considerably below the range found for superheated ammonia in practice, and it is believed that the considerations appearing in the discussion of the heat equations derived from the characteristic equation justify the use of equation [18] in preference to Wobsa's equations.

57 In Fig. 6 the isotherms deduced from equation [18] are drawn on the $pv-p$ plane and the various experimental points are plotted

in order to show the agreement. From this chart it is seen that if both of the determinations of Perman and Davies at 32 deg. are correct, the one at one atmosphere and the other at one-half atmosphere, then either the saturation curve should lie considerably higher or the 32 deg. isotherm should be sharply curved. Since one atmosphere

TABLE 3 SUMMARY OF VARIOUS DETERMINATIONS OF VOLUME OF THE SUPERHEATED VAPOR

Pressure, Atmos.	Temperature, Deg. Cent.	Volume in Liters per Gram	Temperature, Deg. Fahr.	Volume, Cu. Ft. per Lb.	Authority
1/2	0	2.6096	32	41.8006	
1	−20	1.19575	−4	19.1535	
	0	1.2973	32	20.7802	Perman
	50	1.5473	122	24.7847	
	100	1.7964	212	28.7747	
	0	1.2955	32	20.7513	Le Duc
	0	1.2974	32	20.7818	Guye

TABLE 4 COMPARISON OF VALUES OBTAINED FROM VARIOUS EQUATIONS FOR VOLUME OF SUPERHEATED VAPOR WITH EXPERIMENTAL VALUES

Pressure, Atmos.	Temperature, Deg. Fahr.	Experimental Values		Computed from 18		Computed by Wobsa I		Computed by Wobsa II	
		Authority	Value	Value	% Diff. from Exp.	Value	% Diff. from Exp.	Value	% Diff. from Exp.
1	−4	Perman	19.154	19.093	−0.32	19.121	−0.17	19.140	−0.07
	32	Perman	20.780	20.769	−0.05	20.745	−0.17	20.767	−0.06
		Guye	20.782	20.769	−0.06	20.745	−0.18	20.767	−0.07
		Le Duc	20.751	20.769	+0.09	20.745	−0.03	20.767	+0.08
	122	Perman	24.785	24.798	+0.05	24.746	−0.16	24.775	−0.04
	212	Perman	28.775	28.730	−0.16	28.695	−0.28	28.725	−0.17
1/2	32	Perman	41.801	41.916	+0.28	41.821	+0.05	41.939	+0.33

determination agrees closely with the determinations of LeDuc and Guye, since there is good authority for the location of the saturation curve, and since the flatness of the isotherms is fairly well established, it would seem that the half atmosphere point of Perman and Davies is probably in error.

58 In Fig. 6 the full line curve represents values resulting from equation [18] and the dotted line curve represents the values resulting from equation [15]. A comparison of Wobsa's equations is not given because the constants in those equations were determined on the assumption of a different pressure-temperature relation along the saturation curve; the substitution in Wobsa's equations of the

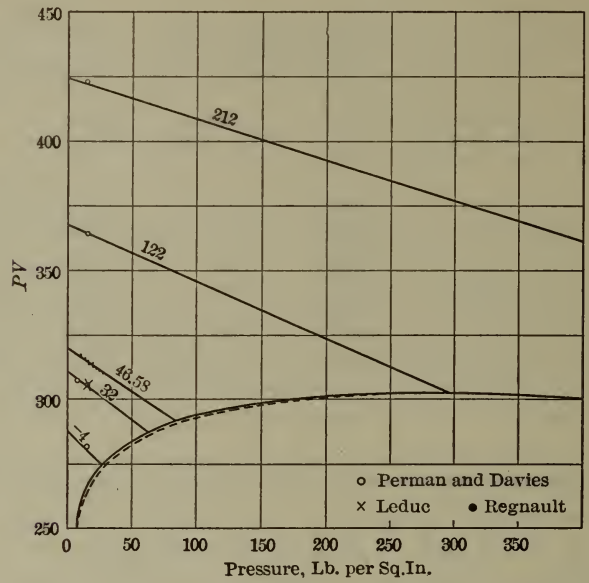

FIG. 6 COMPARISON OF ISOTHERMS DEDUCED FROM EQUATION [18] WITH POINTS REPRESENTING EXPERIMENTAL DETERMINATIONS SHOWN ON $pv-p$ PLANE

values of pressure corresponding to given temperatures as used in this investigation would not therefore give values of volume comparable with those deduced from equation [18]. It may be stated, however, that Wobsa's equations give very fair agreement with the values found experimentally by Dieterici, the maximum deviation of the second equation from the values given in Dieterici's table being a little over 2 per cent.

59 In addition to the preceding the only data available for the purpose of checking the characteristic equation are the results of a series of experiments performed by Regnault. In these experiments measurements were made of the relative volumes occupied by a

quantity of ammonia gas at different pressures along the isotherm corresponding to a temperature of 8.1 deg. cent. or 46.58 deg. fahr. As the weight of ammonia used was not recorded, the experiments can only be used to obtain the relative values of the specific volume or of the product pv along this isotherm. In making use of these values the products pv were plotted against the corresponding values of p, both quantities being measured in the units employed by Regnault. A straight line was next passed through these points by the method of least squares. It was found that the pressure equivalent to 20 lb. per sq. in. came at about the center of the group of points, and the value of the specific volume was calculated by equation [18] for this pressure and a temperature of 46.58 deg. fahr.; this value multiplied by the pressure gave a value of the product pv in English units. The value of this product in the units used by Regnault was then found at the point where the straight line determined by least squares crossed the pressure coördinate equivalent to a pressure of 20 lb. per sq. in. From these two values for pv the conversion factor 0.00561 was found, this being the number by which Regnault's values of pv must be multiplied in order that the isotherm calculated from equation [18] shall pass through the center of Regnault's group of points. All of his values of pv were multiplied by this factor and the points plotted in Fig. 6 as small black dots. The only information they afford is a check on the slope of the isotherm for 46.58 deg. fahr. on the $pv-p$ plane. The points seem to indicate that the isotherms near this temperature at least, are very flat curves or even straight lines.

60 *Specific Heat of the Superheated Vapor.* The first determination of the specific heat of superheated ammonia was made by Regnault, who found a value of 0.50836 at atmospheric pressure and over a temperature range of 75 deg. fahr. to 420 deg. fahr. Later Wiedemann performed two series of experiments at a pressure of about 16 lb. per sq. in.; he found a value of 0.5202 between 77 and 212 deg. fahr. and a value of 0.5356 between 77 and 392 deg. fahr. He proposed the equation

$$c_p = 0.4949 + 0.000172\ t$$

61 Thus far the history is much the same as in the case of superheated steam, the specific heat of which was long supposed to be independent of both temperature and pressure and equal to 0.48 as determined by Regnault; later the investigations of Maillard and LeChatelier and of Lange agreed in making it a linear function of

the temperature. In the case of steam, however, the experiments of Knoblauch and Jakob, Thomas, and Knoblauch and Mollier have shown that the specific heat of superheated steam depends upon the pressure. While no experiments have been made to investigate the variation of the specific heat of superheated ammonia with pressure, it is very probable that there is such a variation and its rate may be determined by means of Clausius' thermodynamic relation providing a sufficiently accurate characteristic equation has been determined. The process of obtaining an explicit expression for c_p in terms of the variables p and T has inherent difficulties and is rendered more difficult in this case by the meagerness of the data available. The success with which the method has been applied to superheated steam by Goodenough, however, seems to warrant its use in the present investigation.

62 The application of the Clausius relation to equation [18] readily gives the following equation for c_p:

$$c_p = \phi\,(T) + \frac{Amn\,(n+1)}{T^{n+1}}\,p \qquad [19]$$

For the complete derivation of this equation see Appendix No. 2.

63 If the arbitrary function $\phi\,(T)$ can be determined, there will result an explicit expression for c_p in terms of the variables p and T. Wobsa has used the same method of analysis starting with his own characteristic equations and has obtained equations for c_p similar to the present one.

64 The determination of the proper function $\phi\,(T)$ is the critical point of the investigation. Wobsa equated his expression for c_p at atmospheric pressure to Wiedemann's linear equation for c_p at atmospheric pressure and thus found the $\phi\,(T)$ to be of the form

$$\phi\,(T) = a + bt - \frac{c}{T^3}$$

He shows that the terms in the third power of T, which is the correction term for atmospheric pressure, is vanishingly small above 212 deg. fahr. and that above this temperature c_p at infinitely small pressures will not differ from c_p at atmospheric pressure, or $\phi\,(T) = 0.4949 + 0.000172t$. Below 212 deg. fahr. where the correction term becomes appreciable, Wobsa substitutes the linear function $\phi\,(T) = 0.4712 + 0.000278\ t$.

65 For the determination of $\phi\ (T)$ in the present investigation there are available the recent determinations of c_p at high temperatures by Nernst. In addition to the results of his own work, Nernst gives a summary of the results of previous investigations (see Table 5). In this table the values are given as molecular specific heats and these have been converted into specific heats per unit of weight by dividing by the molecular weight of NH_3 or 17.064. It is seen that the values given for Wiedemann are lower than those quoted earlier in this section from the original; while the value given for Regnault is higher than the original. As the original papers of Keutel and

TABLE 5 SUMMARY OF C_p MEASUREMENTS AS GIVEN BY NERNST

Temperature, Deg. Cent.	Molecular C_p	Temperature, Deg. Fahr.	C_p	Authority
20	8.64	68	0.506	Keutel
20	8.62	68	0.505	Voller
25–100	8.84	77–212	0.518	E. Wiedemann
25–200	9.11	77–392	0.534	E. Wiedemann
24–216	8.71	75–421	0.510	Regnault
365–567	10.4	689–1053	0.609	Nernst
480–680	11.2	896–1256	0.656	Nernst

Voller are not at hand the values given in this table for their results are assumed to be correct, but the values used for Wiedemann and Regnault are those taken from the original.

66 If in the equation for c_p, $p=0$, the equation reduces to $c_p=\phi\ (T)$. Therefore if we have values for c_p at zero pressure the $\phi\ (T)$ may be determined. To obtain these values the proper correction terms were calculated and subtracted from the values of c_p. The resulting values of $(c_p)_o$ seemed to justify the assumption that $(c_p)_o$ is a linear function of the temperature. This linear relation may not, and probably does not, hold for the entire range of superheat, but it may be assumed as a close approximation, and is the only assumption justified by the experimental data available. The function therefore takes the simple form

$$\phi\ (T)=\alpha+\beta T$$

and the equation for c_p becomes

$$c_p=\alpha+\beta T+\frac{Amn\ (n+1)}{T^{n+1}}p \qquad [20]$$

67 The constants m and n are those heretofore given in connection with the characteristic equation, and the values finally chosen for α and β are: $\alpha=0.382$; $\beta=0.000174$. Replacing $Amn\ (n+1)$ by a single constant C, and substituting the proper value for the various constants the formula for the specific heat becomes:

$$C_p=0.382+0.000174\ T+p\frac{C}{T^6} \quad \ldots\ldots\ldots\ldots [21]$$

where $\log C=13.644705$.

68 Values of c_p have been calculated for various pressures up to 400 lb. per sq. in. and for temperatures up to 1100 deg. fahr.; these values are plotted against temperatures in Fig. 7. The experimental determinations are also plotted to show the agreement. The determinations of Regnault, Nernst, Keutel and Voller were made at atmospheric pressure, while those of Wiedemann were at a pressure of about 16 lb. per sq. in. absolute. The agreement between the curve derived from equation [21] for a pressure of 15 lb. per sq. in. and the points representing the determinations of Regnault, Nernst, Keutel and Voller is seen to be very good, while the points representing Wiedemann's determinations lie considerably above the curve.

69 *Heat Content of Saturated and Superheated Vapor.* Having an explicit formula for the specific heat at constant pressure in terms of the variables p and T, an expression for the heat content in terms of these variables may be easily derived:

$$i=\alpha T+\frac{\beta}{2}T^2-A(n+1)p\frac{m}{T^n}-Acp+i_o \quad \ldots\ldots\ldots [22]$$

For complete derivation of this formula, see Appendix No. 3.

70 The constant of integration i_o is determined by passing to the saturation limit. Since the constants α, β, m, n and c are known, the value of the right-hand member of equation [22], exclusive of i_o may be found for any given pressure and temperature. If the value of i is known at this point, i_o may be determined by subtraction. Now for the saturated vapor

$$i''=i'+r=u+Apv'+r$$

If the energy u be taken equal to zero at 32 deg. fahr., then at this temperature

$$i''_{32}=0+Ap_{32}\,v'_{32}+r_{32}$$

Substituting the proper values as determined by the formulae for the saturated vapor there results

$$i''_{32} = 0.3 + 546.4 = 546.7$$

By the substitution of the proper values of p and T in equation [22] there results

$$i = i_o = 188.7$$

By subtraction $i_v = 358.0$

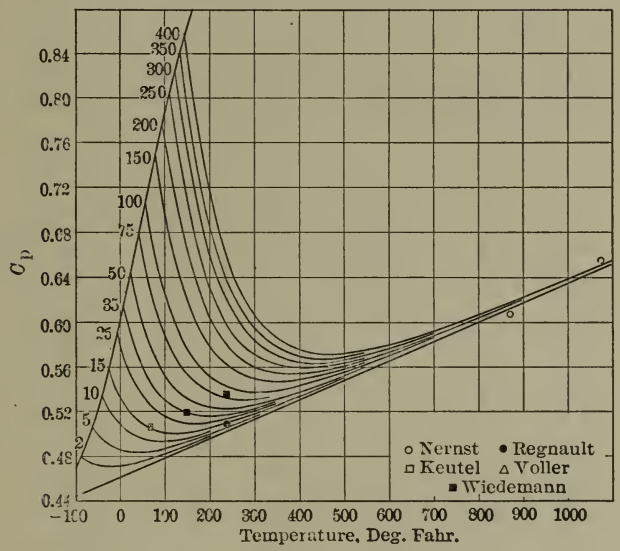

FIG. 7 CURVES GIVING VALUES OF SPECIFIC HEAT AT CONSTANT PRESSURE

Introducing known constants equation [22] becomes, with pressure in lb. per sq. in.

$$i = 0.382T + 0.000087T^2 - p\frac{C}{T^5} - 0.0185p + 358.0 \ldots\ldots [23]$$

where $\log C = 12.945735$.

71 There are no direct measurements of the total heat or heat content of the superheated or saturated vapor by which equation [23] may be checked. A check on the accuracy of the equation at the saturation limit is afforded by a comparison of the values of the heat content of the liquid, obtained by subtracting values of r calculated by equation [14] from values of i'' calculated by equation

[23], with the experimental determinations of Dieterici. This comparison is shown in the following section. A check on the accuracy of the formula in the superheated region may be obtained by a comparison, by means of the law of corresponding states, of the values for the Joule-Thomson coefficient resulting from equation [23] with the values of this coefficient for other vapors. There is also available

TABLE 6 HEAT CONTENT OF LIQUID AS DEDUCED FROM EXPERIMENTS OF DIETERICI AND DREWES

C'im ($t-0°$ C)	u' Above 0°C. Cal. of 4.222Joules	u' Above 0°C. Cal. of 4.184Joules	Temperature, Deg. Cent.	Temperature, Deg. Fahr.	u' Above 32° F. in B.t.u.	$i'=u'+Ap v'$	Authority
1.148	10.65	10.75	9.28	48.70	13.95	19.77	
1.162	10.93	11.03	9.41	48.94	19.85	20.27	
1.141	24.25	24.47	21.25	70.25	44.04	44.67	
1.140	24.57	24.79	21.55	70.79	44.62	45.26	Dieterici
1.139	38.67	39.02	33.95	93.11	70.23	71.21	
1.147	39.27	39.62	34.24	93.63	71.31	72.30	
1.200	42.03	42.41	35.02	95.03	76.34	77.35	
1.172	48.17	48.60	41.1	105.96	87.46	88.68	
1.158	48.35	48.78	41.75	107.15	87.78	89.02	
1.186	61.44	62.00	51.8	125.22	111.60	113.25	Drewes,—
1.163	60.42	60.97	51.95	125.46	109.75	111.41	reliable
1.187	70.74	71.38	59.6	139.3	128.50	130.57	according
1.164	72.75	73.41	62.5	144.5	132.15	134.40	to Diet-
1.179	82.76	83.50	70.2	158.4	150.30	153.07	erici
1.205	87.48	88.27	72.6	162.7	158.90	161.84	
0.974	11.50	11.60	11.8	53.24	20.89	21.35	
1.098	21.62	21.81	19.7	67.46	39.26	39.86	Drewes,—
1.069	22.82	23.02	21.35	70.43	41.44	42.08	ques-
1.140	34.61	34.92	30.36	86.64	62.85	63.72	tioned by
1.148	35.83	36.16	31.21	88.17	65.08	65.97	Dieterici
0.7065	−10.49	−10.59	−14.85	5.27	−19.06	−18.91	

a set of throttling experiments performed by Wobsa. This check is given in Appendix No. 4 and Fig. 10.

72 *Heat Content of the Liquid.* The values of heat content of the liquid may be found by subtracting the values of r found by equation [14] from the values of i'' found by equation [23]. The most important check on these values is derived from a series of experiments performed by Dieterici and by Drewes working in Dieterici's laboratory. In these experiments the quantity measured was the amount of heat given up by a certain weight of a mixture of saturated and liquid ammonia cooling at constant volume in a sealed tube.

This quantity was corrected for the heat due to the latent heat of the portion of vapor condensed and the remaining amount of heat was divided by the change in temperature to obtain the mean "inner" specific heat of the liquid over that temperature range. In his paper Dieterici gives only the formula which he chose as best representing the results of his experiments, which states that the mean "inner"

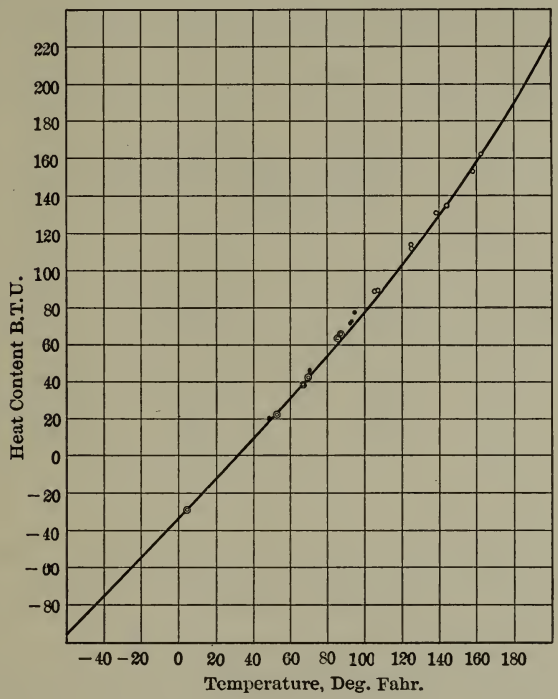

FIG. 8 COMPARISON OF CALCULATED VALUES OF HEAT CONTENT OF LIQUID WITH VALUES DEDUCED FROM EXPERIMENTAL WORK OF DIETERICI AND DREWES

specific heat between t and 32 deg. fahr. $= 1.118 + 0.000578\ (t-32)$. Since this is a linear function of the temperature it follows that the instantaneous "inner" specific heat $= 1.118 + 0.001176\ (t-32)$. The original data of the experiments are not given, but Professor Dieterici has kindly furnished the values which he actually obtained for the mean "inner" specific heat and also a copy of Mr. Drewes' dissertation containing the values obtained by him. If each of these values of the mean "inner" specific heat is multiplied by the temperature

range covered the result will be approximately equal to the internal energy u above 32 deg. fahr. at the temperature t. Since $i' = u' + Apv'$, in order to obtain values of i' the product Apv' corresponding to the temperature t must be added to the value of u' obtained as described. The results of these calculations appear in Table 6. Since Dieterici reported his results in terms of a mean calorie equal to 4.222 joules while in the present investigation a mean calorie of 4.184 joules is used, all of Dieterici's heat quantities must be multiplied by 1.009. The table includes the determinations made by Dieterici himself and all of the determinations made by Drewes; part of the latter are called reliable by Dieterici, while the rest are marked "falsch" in

TABLE 7 SUMMARY OF DETERMINATIONS OF THE SPECIFIC HEAT OF LIQUID AMMONIA

Temperature, Deg. Cent.	Temperature, Deg. Fahr.	Specific Heat of Liquid	Authority
62–30	144–86	1.2286	von Strombeck
20–0	68–32	1.02	Elleau and Ennis
77.7	139.9	0.886	Ludeking and Starr
20–16	68–61	1.094	A. J. Wood

the copy of Drewes' dissertation loaned by Dieterici. The ground for this statement is not given. In Fig. 8 Dieterici's values are plotted as black dots, and those of Drewes as circles. The values questioned by Dieterici are indicated by double circles. The curve represents the values found by the use of equations [14] and [23]. It is seen that in several cases the curve passes between the Dieterici points and the doubtful Drewes points.

73 Since the quotient obtained by dividing the change in i' between two temperatures by the difference in temperature is very nearly equal to the specific heat, the other determinations which have been made of the specific heat may be used to check the calculated values of i'. Table 7 contains a summary of the determinations of the specific heat of the liquid in addition to the work of Dieterici and Drewes already given; this includes the work of Elleau and Ennis, von Strombeck, Ludeking and Starr and A. J. Wood. In Fig. 9 these values are plotted, together with the curves representing the following equations used in the computation of various tables:

Dieterici ..... $c' = 1.118 + 0.001156\ (t-32)$
Zeuner ..... $c' = 1.01235 + 0.00468\ (t-32)$
Wood ..... $c' = 1.12136 + 0.000438t$
Ledoux ..... $c' = 1.0058 + 0.0020322\ (t-32)$
Elleau and Ennis ..... $c' = 0.9834 + 0.0020322\ (t-32)$
Peabody ..... $c' = 1.1$ constant

74 The full line curve represents the values of the tangents to the heat content curve of the present investigation. Since according

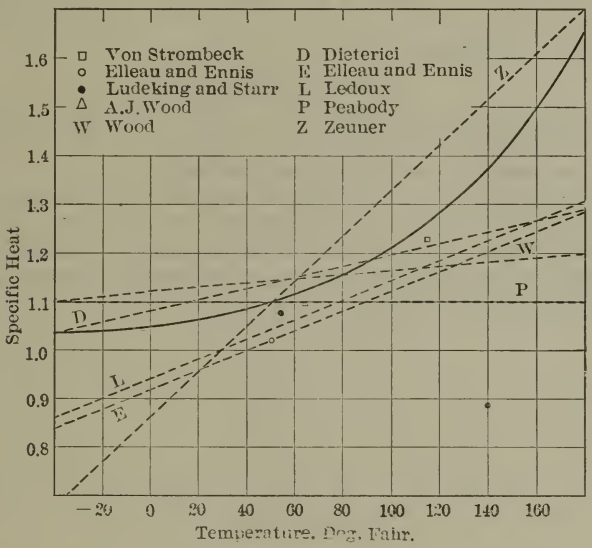

Fig. 9 Comparison of Various Determinations of Specific Heat of Liquid and Various Equations proposed to Represent Same Quantity

to modern ideas of the critical point the specific heat of the liquid there becomes equal to plus infinity, the full line curve is of a more rational form than any linear relation.

75 *Entropy of the Saturated and Superheated Vapor.* An expression for the entropy is readily derived from the characteristic equation and the expression for c_p:

$$s = a \log_e T + \beta T - AB \log_e p - Anp \frac{m}{T^{n+1}} + S_o \ldots\ldots\ldots [24]$$

For the complete derivation of this equation, see Appendix No. 3.

76 The constant s_o is determined by passing to the saturation limit at 32 deg. fahr., where $s'=0$ and therefore $s''=\frac{r}{T}$. Substituting this value for s in the left-hand member of equation (s) and the proper values of p and T in the right-hand member, s_o is found to be equal to -0.82656. Substituting this value and the known values of the other constants, and passing to common logarithms equation [24] becomes

$$s=0.8796 \log T+0.000174\ T-0.2695 \log p-\frac{p}{T^6}C-0.82656 \quad [25]$$

where $\log C=12.866554$.

ACKNOWLEDGMENT

77 Acknowledgment is hereby made to Prof. G. A. Goodenough of the University of Illinois, who suggested this investigation and who has given to it his hearty encouragement and his personal supervision.

78 The investigation was made under the auspices of the mechanical engineering department of the Engineering Experiment Station of the University of Illinois, and complete tables based upon the formulae set forth in this paper, together with a Mollier diagram and a full bibliography of the subject will in due time be published as a bulletin of the station.

APPENDIX No. 1

METHOD OF CONSTRUCTION OF FIG. 1

79 Using formula [7] to obtain the saturation temperatures of ammonia from those of water

$$\frac{1}{T_a}=c\frac{1}{T_w}+k$$

Let

$$\frac{-1}{T_w}=y \quad \text{and} \quad \frac{-1}{T_a}=x$$

then

$$-x=-cy+k$$

Since y is a function of the water vapor temperature corresponding to pressure p, y is also a function of p; the value of y for a given value of p may be found by making use of the tables of water vapor pressure; x is a function of the temperature of the ammonia corresponding to this same vapor pressure p; the curve giving x as a function of y is evidently a straight line.

80 Since y is a function of the pressure p and x a function of the corresponding temperature, the values of x and y may represent their respective values of pressure and temperature. The corresponding values of saturation temperature and vapor pressure may be read directly from the diagram.

81 Fig. 1 was constructed in the following manner: Integral values of temperature were assumed and the corresponding values of x were computed according to the relation $x=-\frac{1}{T}$. These x's were then laid off to a convenient scale and each labeled with the temperature to which the value of x corresponds. Integral values of pressure were assumed, the corresponding values of temperature found from steam tables, and the corresponding values of y computed according to the relation $y=-\frac{1}{T}$. These y's were then laid off to a convenient scale and each labeled with the pressure to which the value of y corresponds.

82 The value of vapor pressure at a given temperature as taken from the Marks and Davis steam tables was found to agree with the value given by the new Marks formula with sufficient accuracy up to a temperature of 400 deg. fahr.; above this temperature the values differed materially. In the construction of the chart, therefore, the steam tables were used to find the pressures corresponding to temperatures below 400 deg. fahr. and the newer formula was used to find the pressures corresponding to temperatures above that point.

83 Table 8 shows the calculations for a few of the values of Fig. 1.

84 In order to settle upon the most probable location of the straight line representing the pressure-temperature relation of saturated ammonia upon this

TABLE 8 SAMPLE CALCULATIONS OF THE COÖRDINATES OF FIG 1

ABSCISSAE

Temperature, Deg. Fahr.	Absolute Temperature	Reciprocal of Absolute Temperature	Corresponding Value of X, In., referred to the point −10 as Origin $= -(8000 \times \pm -10)$
−100	359.64	0.0027806	12.244
−50	409.64	0.0024412	9.529
0	459.64	0.0021756	7.405
+50	509.64	0.0019622	5.697
100	559.64	0.0017869	4.295
150	609.64	0.0016403	3.122
200	659.64	0.0015160	2.128
250	709.64	0.0014092	1.273

ORDINATES

Pressure, Lb. per Sq. In.	Corresponding Saturation Temperature of Water Vapor, Deg. Fahr.	Absolute Temperature	Reciprocal of Absolute Temperature	Corresponding Value of y, In., referred to the point −18 as Origin $= -(20000 \times \pm -18)$
1	101.83	561.47	0.0017810	17.621
5	162.28	621.92	0.0016079	14.158
10	193.22	652.86	0.0015317	12.634
50	281.0	740.64	0.0013502	9.004
100	327.8	787.44	0.0012699	7.399
500	467.2	926.84	0.0010789	3.579
1000	544.9	1004.54	0.0009955	1.910
1700	613.4	1073.04	0.0009319	0.639

TABLE 9 SUMMARY OF DETERMINATIONS OF CRITICAL DATA FOR AMMONIA

Investigator	Date	Temperature, Deg. Cent.	Pressure, Atmospheres	Temperature, Deg. Fahr.	Pressure, Lb. per Sq. In.
Dewar.....	1884	130.0	115	266.0	1690.0
Vincent and Chappuis..	1886	131.0	113	267.8	166.08
Jaquerod.....	1908	132.3	109.6	270.1	1610.7
Sheffer.....	1910	132.1	[111.3]	269.8	1635.7

chart the experimental data must be plotted. The pressures observed by Pictet over a temperature range of −22 to 122 deg. fahr. agree almost exactly with those observed by Regnault; they are neither tabulated nor plotted. Faraday's results are quite inconsistent with those of Regnault, and are neither plotted in Fig. 1 nor given weight in the determination of the constants of the equation.

85 The two points in the upper right-hand corner of Fig. 1 represent the experiments at the critical point. The open square shows the values given by Vincent and Chappuis and the solid triangle the values given by Dewar. The values from which these points were plotted are given in Table 9.

86 This table also contains the values for critical data given by Jaquerod and Scheffer, which were found after the plate for Fig. 1 was made.

APPENDIX No. 2

DERIVATION OF EQUATION [19]

$$c_p = \phi T + \frac{Amn\,(n+1)}{T^{n+1}}\,p$$

87 The Clausius relation is expressed by the equation

$$\left[\frac{\partial c_p}{\partial p}\right]_T = -AT\left[\frac{\partial^2 v}{\partial T^2}\right]_p$$

The derivative in the right-hand member is determined from the characteristic equation. Thus from equation [18]

$$\frac{\partial v}{\partial T} = \frac{B}{p} + \frac{mn}{T^{n+1}} \qquad [26]$$

and

$$\frac{\partial^2 v}{\partial T^2} = -\frac{mn\,(n+1)}{T^{n+2}} \qquad [27]$$

Substituting the second derivative in the Clausius relation the result is

$$\left[\frac{\partial c_p}{\partial p}\right]_T = \frac{Amn\,(n+1)}{T^{n+1}} \qquad [28]$$

Taking T as a constant and integrating with respect to p as the independent variable, the result is

$$c_p = \frac{Amn\,(n+1)}{T^{n+1}}\,p + \text{constant of integration}$$

The constant of integration may be a function of T, since T was held constant during the integration; hence

$$c_p = \phi\,(T) + \frac{Amn\,(n+1)}{T^{n+1}}\,p \qquad [29]$$

APPENDIX No. 3

DERIVATION OF EQUATIONS [22] AND [24]

$$i=\alpha T+\frac{\beta T^2}{2}-A\ (n+1)\ p\ \frac{m}{T^n}-Ac_p+i_o$$

$$s=\alpha\ \log_e T+\beta T-AB\ \log_e p-Anp\ \frac{m}{T^{n+1}}+s_c$$

88 Having an explicit formula for the specific heat at constant pressure n terms of the variables p and T, an expression for the heat content in terms c. these variables may be easily derived. For this purpose the general equation

$$dq=c_p dT-AT\left[\frac{\partial v}{\partial T}\right]_p dp \qquad [30]$$

is most convenient. Since the heat content is defined by the relation

$$i=A\ (u+pv)$$

then

$$di=A\ [du+d(pv)]$$

or

$$di=dq+Avdp$$

Hence by substitution in equation [30] there results

$$di=c_p dT-A\left[T\frac{\partial v}{\partial T}-v\right]dp \qquad [31]$$

From the characteristic equation

$$\frac{\partial v}{\partial T}=\frac{B}{p}+\frac{nm}{T^{n+1}}$$

whence

$$T\frac{\partial v}{\partial T}-v=(n+1)\ \frac{m}{T^n}$$

Substituting this and the general expression for c_p in equation [31] the result is

$$di=(\alpha+\beta T)\ dT+Amn\ (n+1)\ p\frac{dT}{T^{n+1}}-\frac{Am\ (n+1)}{T^n}\ dp-Acdp \qquad [32]$$

Since i depends upon the state of the substance only, the second member of equation [30] must be an exact differential. The integral is readily found to be

$$i = \alpha T + \frac{\beta}{2} T^2 - A(n+1)\frac{pm}{T^n} - Acp + i_o \quad [33]$$

89 An expression for entropy is readily found from equation [30]. Thus

$$ds = \frac{dq}{T} = c_p \frac{dT}{T} - A\left[\frac{\partial v}{\partial T}\right]_p dp \quad [34]$$

Introducing in equation [34] the expressions previously derived for c_p and $\left[\frac{\partial v}{\partial T}\right]_p$, the result is

$$ds = \left[\frac{\alpha}{T} + \beta\right] dT + Amn\,(n+1)p\frac{dT}{T^{n+2}} - \frac{ABdp}{p} - \frac{Amn}{T^{n+1}}dp \quad [35]$$

The integral of equation [35] is

$$s = \alpha \log_e T + \beta T - AB \log_e p - Anp\frac{m}{T^{n+1}} + s_o \quad [36]$$

APPENDIX No. 4

TEST OF VALIDITY OF EQUATION [23] BY MEANS OF THE LAW OF CORRESPONDING STATES

90 Since i is constant in a throttling process, the Joule-Thomson coefficient μ may be defined as the derivative

$$\left[\frac{\partial T}{\partial p}\right]_i$$

From calculus

$$\left[\frac{\partial T}{\partial p}\right]_i = -\frac{\dfrac{\partial i}{\partial p}}{\dfrac{\partial i}{\partial T}}$$

and from the definition of heat content i

$$\frac{\partial i}{\partial T} = c_p$$

Hence

$$\mu = \left[\frac{\partial T}{\partial p}\right]_i = -\frac{1}{c_p} \cdot \frac{\partial i}{\partial p}$$

or

$$\mu = \frac{A}{c_p}\left[\frac{m\,(n+1)}{T^n} + c\right] \qquad [37]$$

91 From equation [37] it is seen that μ varies with the pressure; as the temperature rises, however, the influence of pressure decreases. Joule and Kelvin, working with gases far removed from the saturation limit, found that μ varies inversely as the square of the absolute temperature. The experimental values of μ for steam expressed in pounds per square inch and degrees fahrenheit were reduced by multiplying by 2.56, a factor which is the ratio of the critical pressure of water (2947 lb. per sq. in.) to the critical temperature (1149 deg. fahr.), these critical values being the ones determined by Cailletet and Colardeau. Davis gives a curve which he says represents the experimental values for water in the best possible manner; the values in this curve are those translated back to ordinary units. Later determination of the critical constants for water by

Holborn and Bauman give 3200 lb. per sq. in. for the critical pressure and 1166 deg. fahr. for the critical temperature Table 10. If then the values of μ on the

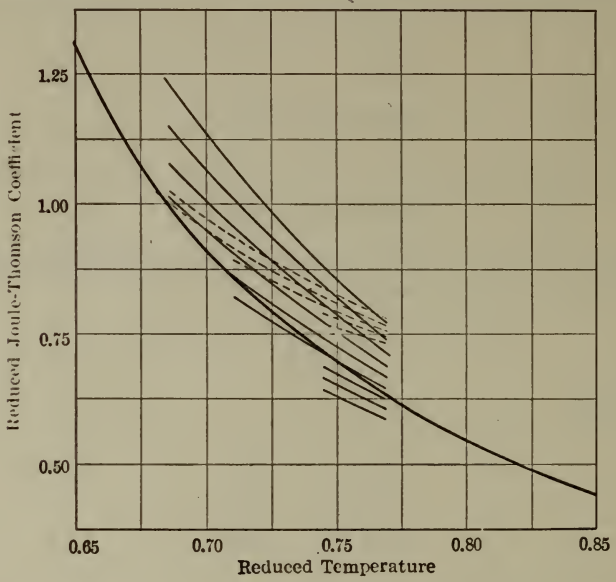

FIG. 10 COMPARISON OF VALUES OF REDUCED JOULE-THOMSON COEFFICIENT FOR AMMONIA ACCORDING TO WOBSA AND VALUES RESULTING FROM EQUATION [37] WITH VALUES FOR WATER AS GIVEN BY DAVIS

TABLE 10 SUMMARY OF DETERMINATIONS OF CRITICAL DATA FOR WATER

Investigator	Date	Critical Temperature, Deg. Fahr.	Critical Pressure, Lb. per Sq. In.
Nadejdine	1885	676.6	.....
Battelli	1890	687.7	2859
Cailletet and Colardeau	1891	689.0	2994
Strauss	1892	698.0	2873
Traube and Teichner	1904	705.2	....
Holborn and Bauman	1910	706.3	3200

Davis curve are multiplied by $\frac{3200}{1166}$, or 2.745, and the temperatures are divided by 1166, there results a curve expressing the variation of μ with temperature, both expressed in reduced units, according to the best available experimental data.

92 This curve is shown in Fig. 10 as the heavy full line curve. The values resulting from [37] and those resulting from Wobsa's equation for ammonia may now be compared with the values for water. Wobsa's throttling experiments were performed at pressures of from 2 to 10 atmospheres and at temperatures of 42.8, 60.8, 86 and 104 deg. fahr. The values of μ at the same pressures and temperatures have been calculated from [37] and are shown in Fig. 10 as full line curves, while the values resulting from Wobsa's equation are shown as dotted curves. It is seen that while the full line curves form a rather strongly diverging brush, yet the Davis line passes through the center of the brush and it appears that the brush would converge into the Davis curve at high temperatures. On the other hand the brush of dotted curves has an entirely different trend and would not merge into the Davis curve.

THE PHYSICAL PROPERTIES OF ANHYDROUS AMMONIA

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ABSTRACT OF PAPER

This paper presents the results of a critical examination of existing information on the physical properties of anhydrous ammonia. That information is of three kinds:

a Direct experimental data

b Formulae based upon experimental data usually in conjunction with certain assumptions

c Tables based upon experimental data and on the formulae

2 All the experimental observations of any value are given in the paper, and those equations which seem to deserve consideration are examined in the light of the experimental evidence. The conclusion reached with reference to the accuracies of all the equations examined, permits the calculation of a table, which probably represents our knowledge of the physical properties of anhydrous ammonia as accurately as any that can be constructed at the present time.

3 *Pressure-Temperature Relation of the Saturated Vapor.* The important experimental work in this field is by Regnault, Pictet, Blumcke, Brill, Davies, and Scheffer; in addition there are boiling-point determinations by de Forcrand, Gibbs, Franklin, Dickerson, and others, and critical-point determinations by Dewar, Vincent and Chappuis, Guye and Scheffer. The observations of Pictet and

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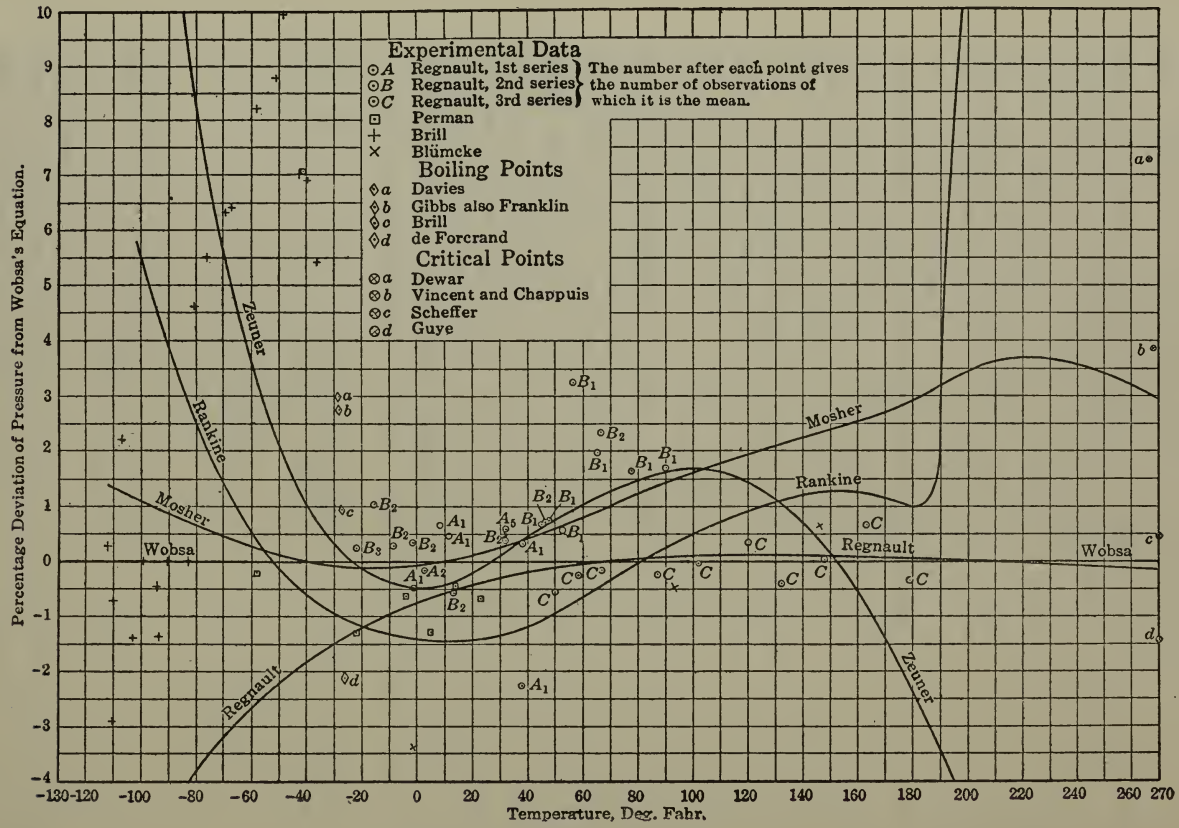

FIG. 1 PERCENTAGE DEVIATION OF PRESSURES DETERMINED EXPERIMENTALLY; AND ALSO OF CERTAIN CALCULATED VALUES OF THE PRESSURE OF SATURATED AMMONIA VAPOR, FROM THE PRESSURES DEDUCED FROM WOBSA'S EQUATION

Scheffer are in close agreement with those of Regnault. All the other observed pressures are plotted in Fig. 1 as percentage deviations from the pressures calculated from an equation proposed by Wobsa which is of the form

$$\log_{10} p = a - ba^t - c\beta^t$$

The constants have the following values:

	English Units Lb. per Sq. In. and Deg. Fahr.	Metric Units Mm. of Mercury and Deg. Cent.
a	9.7947	11.5083
$\log_{10} ba^t$.....	$0.8704939 - 0.0002214t$	$0.863408 - 0.0003986t$
$\log_{10} c\beta^t$.....	$-0.04742 \quad -0.0033055t$	$-0.15319 \quad -0.00595t$

4 In the same figure are shown, in the same manner, the pressures calculated from other vapor-pressure equations. It will be seen that the Wobsa equation represents the experimental data more accurately than any of the other equations throughout the whole range from the freezing temperature (−108 deg. fahr.) to the critical temperature (270 deg. fahr.). The Regnault equation, which has been most used, becomes quite inaccurate at low temperatures. The Mosher equation gives more weight to the observations of Regnault's second series (marked *B* in Fig. 1) and neglects the third series points (marked *C*). An inspection of the figure shows that the second series points are not self consistent and are not supported by any other experimental evidence, whereas the third series observations are consistent and are well supported by Blumcke's observations. Moreover, Regnault in his choice of vapor-pressure equations practically disregarded his second series points. The only part of the temperature range in which Wobsa's equation does not represent the experimental data satisfactorily, is from −30 deg. to −80 deg. fahr. The experimental work in this region is by Brill and is open to some suspicion; it is not probable, however, that his work is sufficiently in error to account for the deviations. Consequently, there is still some uncertainty as to the vapor pressures at this part of the range. Throughout the commercially and technically important range from −20 to 150 deg. fahr., the saturation pressures are probably known to within 1 per cent.

5 *Specific Volumes of the Saturated and Superheated Vapor.* No experimental information was available before 1904 on specific volumes, with the exception of a few determinations at 0 deg. cent.

and 760 mm. pressure. Since that time there have been published some determinations of the specific volumes of the saturated vapor by Dieterici, and of the superheated vapor by Perman and Davies. Dieterici's work does not go below 3 deg. cent. and Perman and Davies give only five measurements. In order to deduce the probable value of specific volumes in the unexplored field, it is necessary to have recourse to a *characteristic equation*; that is, an equation relating pressure, volume, and temperature of the vapor.

6 It has been assumed by all investigators that the form of characteristic equation which has been found to represent the properties of steam, will probably be found to be applicable to ammonia and other vapors when the correct constants are inserted. These constants can be determined and the equations can be tested by the use of information other than the direct observations of volumes. There are available for this purpose some isolated determinations of coefficients of compressibility and dilatation by Regnault, Lord Rayleigh, and Leduc; and some Joule-Thomson coefficients determined from the throttling experiments of Wobsa, as well as some other material of problematical value. The direct determinations of latent heat by Regnault and von Strombeck, which might be used for the calculation of the specific volumes of the saturated vapor by means of the well-known Clapeyron equation, are too inconsistent to be of any value.

7 All of this material would be adequate for establishing a characteristic equation if it were reliable. Unfortunately, an examination of the material discloses a number of inconsistencies.

8 In view of the uncertainty as to which of the experimental observations are trustworthy, the best that can be done at present is to select a characteristic equation which shows good general agreement with experiment. It is possible to construct an indefinite number of such equations, but it is not at present possible to establish any one as definitely accurate, or even definitely better than some other one which shows the same kind of general agreement with experimental data. Under these conditions, it has not seemed worth while to devise a new equation since one, at least, of the recent equations shows a satisfactory general agreement with the experimental data and represents the results of experiment as closely as the accuracy of the experimental work warrants.

9 All the tables of the properties of ammonia which were published prior to 1904, including the well-known tables of Ledoux, Zeuner, Wood and Peabody, antedate the experimental determina-

tion of specific volumes and were based upon the scantiest of experimental information and upon certain unverified assumptions. In addition, the forms of the characteristic equations used by these authorities, have been found by recent investigations to be unsatisfactory for steam, and consequently the primary justification for their use has disappeared. It follows that the characteristic equations and the tables of Ledoux, Zeuner, Wood and Peabody, must be regarded as definitely displaced by those equations and tables which are based upon the later experimental information.

10 Four tables of the properties of ammonia have been published during the past eight years. The earliest of these (1904) is by Dieterici and is calculated directly from his own observations without the use of a characteristic equation. It gives no information on the properties of the saturated vapor at temperatures below 32 deg. fahr., nor does it throw any light on the superheated region. The three remaining tables are based upon practically all the information now available and are calculated from characteristic equations of forms recently established as satisfactory for steam. They are the only tables that need be seriously considered. Of these tables, two are by Wobsa and one by Hybl. Wobsa himself decided that his later table was better than the earlier one, and this decision is justified by an examination of the facts. There remain only the later Wobsa and the Hybl tables. A critical examination of the characteristic equations on which these are based, establishes the definite superiority of the Wobsa equation, and also shows an inherent defect in the Hybl equation. It shows, moreover, a general agreement of the Wobsa equation with the experimental data which is probably as good as is possible in view of the inconsistencies in the experimental work. The Wobsa equation is

$$v = 0.0075 + \frac{49.736T}{p} - \frac{2450}{T^2} + \frac{80}{p}$$

where

T = absolute temperature, deg. cent.
v = specific volume, cu. m. per kg.
p = pressure, kg. per sq. m.

11 A further examination of the experimental data indicates the following as the best available equations:
For *the specific heat of the superheated vapor*

$$C_p = 0.505 + 0.00017t + 0.424\frac{t^3}{10^9} -$$

For *the specific heat of the liquid*

$$c = 1.118 + 0.00208t$$

For *the specific volume of the liquid*

$$v = 3.7 - \log_{10}(134 - t)$$

In the above equations t = temperature, deg. cent.

v = specific volume, cc. per gram

12 From the preceding equations, there are deduced the following: The *entropy of saturated or superheated vapor* is

$$N = 0.49 \log_e T - AR \log_e P - \frac{11.5}{T^3} \cdot P - 0.38$$

and *the total heat* is

$$H = 0.49t - \left(\frac{17.22}{T^2} - 0.0000178\right) P + 309.6$$

where

H = total heat, calories

P = pressure, kg. per sq. m.

$R = 49.736$

$A = 0.0023442$

13 Wobsa prepared a table based on the above equations, giving the properties of saturated ammonia from −30 to +40 deg. cent. by 5 deg. intervals: This table has been made use of in the computation of the greater part of Table 1, which gives the properties of saturated ammonia by 1 deg. intervals from −22 deg. to +131 deg. fahr.

TABLE 1 PROPERTIES OF SATURATED AMMONIA VAPOR BASED ON WOBSA'S SECOND EQUATION

Temperature, Deg. Fahr.	Pressure, Lb. per Sq. In. Abs.	Specific Volume, of Liquid, Cu. Ft. per Lb.	Specific Volume of Vapor, Cu. Ft. per Lb.	Density, Lb. per Cu. Ft.	Heat of Liquid, B.T.U.	Total Heat, B.T.U.	Latent Heat B.T.U.			Entropy		
							Total	Internal	External	Liquid n	Vapor N	Difference $N-n$
−22	16.98	0.02368	15.80	0.06329	−58.91	524.8	583.7	534.2	49.50	−0.1265	1.2090	1.3355
−21	17.46	0.02370	15.41	0.06489	−57.84	525.2	583.0	533.4	49.58	−0.1241	1.2068	1.3309
−20	17.95	0.02372	15.03	0.06653	−56.77	525.5	582.3	532.6	49.65	−0.1217	1.2046	1.3263
−19	18.44	0.02374	14.66	0.06821	−55.71	525.8	581.6	531.9	49.73	−0.1193	1.2023	1.3216
−18	18.93	0.02377	14.30	0.06993	−54.64	526.2	580.9	531.1	49.80	−0.1169	1.2001	1.3170
−17	19.43	0.02379	13.95	0.07168	−53.57	526.6	580.2	530.3	49.88	−0.1145	1.1979	1.3124
−16	19.93	0.02381	13.61	0.07348	−52.51	527.0	579.5	529.5	49.95	−0.1121	1.1957	1.3078
−15	20.45	0.02384	13.27	0.07536	−51.45	527.3	578.8	528.8	50.03	−0.1097	1.1935	1.3032
−14	21.00	0.02386	12.93	0.07734	−50.39	527.7	578.1	528.0	50.10	−0.1072	1.1912	1.2984
−13	21.58	0.02389	12.60	0.07937	−49.32	528.1	577.4	527.2	50.18	−0.1048	1.1891	1.2939
−12	22.17	0.02391	12.28	0.08143	−48.25	528.4	576.7	526.4	50.25	−0.1024	1.1870	1.2894
−11	22.76	0.02394	11.97	0.08354	−47.17	528.7	575.9	525.6	50.31	−0.1001	1.1848	1.2849
−10	23.36	0.02396	11.69	0.08554	−46.10	529.1	575.2	524.8	50.38	−0.0977	1.1827	1.2804
− 9	24.96	0.02398	11.41	0.08764	−45.02	529.4	574.4	524.0	50.44	−0.0954	1.1806	1.2760
− 8	24.56	0.02401	11.13	0.08985	−43.95	529.7	573.7	523.2	50.51	−0.0930	1.1785	1.2715
− 7	25.18	0.02403	10.86	0.09208	−42.87	530.1	572.9	522.3	50.58	−0.0906	1.1763	1.2669
− 6	25.81	0.02405	10.61	0.09425	−41.80	530.4	572.2	521.6	50.64	−0.0883	1.1742	1.2625
− 5	26.46	0.02408	10.37	0.09643	−40.73	530.6	571.4	520.7	50.71	−0.0859	1.1721	1.2580
− 4	27.14	0.02411	10.13	0.09872	−39.65	530.9	570.6	519.8	50.78	−0.0835	1.1699	1.2534
− 3	27.83	0.02413	9.90	0.1010	−38.57	531.3	569.9	519.0	50.84	−0.0811	1.1680	1.2491
− 2	28.53	0.02416	9.68	0.1033	−37.48	531.6	569.1	518.2	50.90	−0.0788	1.1660	1.2448
− 1	29.23	0.02418	9.48	0.1055	−36.40	532.0	568.4	517.4	50.97	−0.0764	1.1641	1.2405
0	29.95	0.02420	9.28	0.1078	−35.32	532.3	567.6	516.6	51.03	−0.0740	1.1621	1.2361
1	30.68	0.02422	9.08	0.1101	−34.24	532.7	566.9	515.8	51.09	−0.0717	1.1602	1.2319
2	31.42	0.02425	8.89	0.1125	−33.15	532.9	566.1	514.9	51.15	−0.0693	1.1582	1.2275
3	32.19	0.02427	8.70	0.1149	−32.07	533.2	565.3	514.0	51.21	−0.0670	1.1563	1.2233
4	32.98	0.02430	8.51	0.1175	−30.99	533.5	564.5	513.2	51.27	−0.0646	1.1543	1.2189
5	33.81	0.02433	8.32	0.1202	−29.90	533.8	563.7	512.4	51.34	−0.0622	1.1524	1.2146
6	34.65	0.02436	8.13	0.1230	−28.81	534.1	562.9	511.5	51.40	−0.0599	1.1505	1.2104
7	35.49	0.02439	7.95	0.1258	−27.71	534.4	562.1	510.6	51.45	−0.0576	1.1486	1.2062
8	36.34	0.02442	7.77	0.1288	−26.62	534.6	561.2	509.7	51.51	−0.0553	1.1466	1.2019
9	37.19	0.02444	7.59	0.1318	−25.52	534.9	560.4	508.8	51.57	−0.0530	1.1447	1.1977
10	38.05	0.02447	7.42	0.1348	−24.42	535.1	559.5	507.9	51.62	−0.0507	1.1428	1.1935
11	38.94	0.02450	7.25	0.1379	−23.33	535.4	558.7	507.0	51.68	−0.0483	1.1409	1.1892
12	39.84	0.02452	7.08	0.1412	−22.24	535.7	557.9	506.2	51.73	−0.0460	1.1390	1.1850
13	40.76	0.02455	6.91	0.1447	−21.14	535.9	557.0	505.2	51.78	−0.0437	1.1370	1.1807
14	41.69	0.02458	6.75	0.1482	−20.04	536.2	556.2	504.3	51.84	−0.0414	1.1351	1.1765
15	42.64	0.02460	6.59	0.1517	−18.93	536.4	555.3	503.4	51.89	−0.0391	1.1331	1.1722
16	43.60	0.02463	6.44	0.1553	−17.82	536.7	554.5	502.5	51.93	−0.0368	1.1312	1.1680
17	44.60	0.02466	6.30	0.1587	−16.71	537.0	553.7	501.7	51.98	−0.0345	1.1292	1.1637

TABLE 1—*Continued*

Temperature, Deg. Fahr.	Pressure, Lb. per Sq. In. Abs.	Specific Volume of Liquid, Cu. Ft. per Lb.	Specific Volume of Vapor, Cu. Ft. per Lb.	Density, Lb. per Cu. Ft.	Heat of Liquid, B.T.U.	Total Heat, B.T.U.	Latent Heat B.T.U.			Entropy		
							Total	Internal	External	Liquid n	Vapor N	Difference $N-n$
18	45.62	0.02469	6.17	0.1621	−15.60	537.2	552.8	500.8	52.03	−0.0322	1.1272	1.1594
19	46.65	0.02472	6.05	0.1653	−14.50	537.5	552.0	499.9	52.07	−0.0299	1.1253	1.1552
20	47.70	0.02474	5.93	0.1686	−13.39	537.7	551.1	499.0	52.11	−0.0275	1.1233	1.1508
21	48.77	0.02477	5.81	0.1721	−12.28	538.0	550.3	498.1	52.16	−0.0252	1.1213	1.1465
22	49.85	0.02480	5.69	0.1756	−11.17	538.3	549.5	497.3	52.21	−0.0229	1.1193	1.1422
23	50.95	0.02483	5.57	0.1795	−10.06	538.5	548.6	496.3	52.25	−0.0206	1.1174	1.1380
24	52.06	0.02485	5.47	0.1828	− 8.94	538.8	547.8	495.4	52.29	−0.0183	1.1156	1.1339
25	53.19	0.02488	5.36	0.1866	− 7.82	539.1	546.9	494.6	52.33	−0.0160	1.1138	1.1298
26	54.36	0.02491	5.25	0.1905	− 6.71	539.3	546.0	493.6	52.38	−0.0138	1.1120	1.1258
27	55.5	0.02494	5.14	0.1946	− 5.59	539.5	545.1	492.7	52.42	−0.0115	1.1102	1.1217
28	56.7	0.02497	5.04	0.1984	− 4.47	539.7	544.2	491.7	52.46	−0.0092	1.1084	1.1176
29	57.9	0.02500	4.94	0.2024	− 3.35	540.0	543.4	490.9	52.50	−0.0069	1.1065	1.1134
30	59.1	0.02502	4.84	0.2066	− 2.23	540.3	542.5	489.9	52.55	−0.0046	1.1047	1.1093
31	60.4	0.02505	4.74	0.2110	− 1.12	540.5	541.6	489.0	52.59	−0.0023	1.1029	1.1052
32	61.7	0.02508	4.64	0.2155	0.00	540.7	540.7	488.1	52.63	0.0000	1.1011	1.1011
33	63.0	0.02511	4.54	0.2203	1.13	540.9	539.8	487.1	52.67	0.0023	1.0993	1.0970
34	64.3	0.02513	4.46	0.2242	2.26	541.2	538.9	486.2	52.71	0.0046	1.0975	1.0929
35	65.7	0.02516	4.37	0.2288	3.39	541.4	538.0	485.2	52.75	0.0069	1.0957	1.0888
36	67.1	0.02519	4.29	0.2231	4.52	541.6	537.1	484.3	52.79	0.0091	1.0939	1.0848
37	68.5	0.02522	4.21	0.2375	5.65	541.9	536.2	483.4	52.83	0.0114	1.0921	1.0807
38	69.9	0.02525	4.13	0.2421	6.78	542.1	535.3	482.4	52.87	0.0136	1.0903	1.0767
39	71.3	0.02528	4.05	0.2469	7.91	542.2	534.3	481.4	52.90	0.0159	1.0885	1.0726
40	72.7	0.02531	3.97	0.2519	9.04	542.4	533.4	480.5	52.94	0.0182	1.0867	1.0685
41	74.1	0.02534	3.89	0.2571	10.17	542.7	532.5	479.5	52.97	0.0205	1.0849	1.0644
42	75.6	0.02537	3.82	0.2618	11.31	542.8	531.5	478.5	53.01	0.0227	1.0831	1.0604
43	77.1	0.02540	3.75	0.2667	12.45	543.0	530.5	477.5	53.04	0.0249	1.0814	1.0565
44	78.6	0.02543	3.68	0.2718	13.60	543.2	029.6	476.5	53.07	0.0272	1.0796	1.0524
45	80.2	0.02546	3.61	0.2770	14.75	543.4	528.6	475.5	53.10	0.0294	1.0778	1.0484
46	81.8	0.02549	3.54	0.2825	15.88	543.5	527.6	474.5	53.13	0.0316	1.0761	1.0445
47	83.4	0.02552	3.47	0.2882	17.02	543.6	526.6	473.4	53.15	0.0338	1.0743	1.0405
48	85.1	0.02555	3.40	0.2941	18.16	543.8	525.6	472.4	53.18	0.0360	1.0726	1.0366
49	86.8	0.02558	3.34	0.2994	19.30	543.9	524.6	471.4	53.20	0.0383	1.0708	1.0325
50	88.5	0.02561	3.28	0.3049	20.45	544.1	523.6	470.4	53.22	0.0405	1.0690	1.0285
51	90.2	0.02564	3.22	0.3106	21.60	544.2	522.6	469.4	53.24	0.0427	1.0673	1.0246
52	92.0	0.02567	3.16	0.3165	22.76	544.3	521.5	468.2	53.27	0.0450	1.0657	1.0207
53	93.8.	0.02570	3.10	0.3226	23.91	544.4	520.5	467.2	53.29	0.0472	1.0640	1.0168
54	95.6	0.02573	3.04	0.3289	25.06	544.6	519.5	466.2	53.31	0.0495	1.0624	1.0129
55	97.4	0.02576	2.98	0.3356	26.22	544.7	518.5	465.2	53.32	0.0517	1.0607	1.0090
56	99.2	0.02579	2.93	0.3413	27.37	544.9	517.5	464.2	53.34	0.0539	1.0590	1.0051
57	101.0	0.02582	2.88	0.3472	28.52	545.0	516.5	463.1	53.36	0.0562	1.0574	1.0012

TABLE 1—*Continued*

Temperature, Deg. Fahr.	Pressure, Lb. per Sq. In. Abs.	Specific Volume, of Liquid, Cu. Ft. per Lb.	Specific Volume of Vapor, Cu. Ft. per Lb.	Density, Lb. per Cu. Ft.	Heat of Liquid, B.T.U.	Total Heat, B.T.U.	Latent Heat B.T.U.			Entropy		
							Total	Internal	External	Liquid n	Vapor N	Difference $N-n$
58	102.9	0.02585	2.83	0.3534	29.67	545.2	515.5	462.1	53.38	0.0584	1.0557	0.9973
59	104.8	0.02589	2.78	0.3597	30.83	545.3	514.5	461.1	53.39	0.0607	1.0540	0.9933
60	106.7	0.02592	2.73	0.3663	31.99	545.4	513.4	460.0	53.41	0.0629	1.0524	0.9895
61	108.6	0.02595	2.68	0.3731	33.16	545.6	512.4	459.0	53.42	0.0651	1.0507	0.9856
62	110.6	0.02599	2.63	0.3802	34.32	545.7	511.4	458.0	53.43	0.0673	1.0491	0.9818
63	112.6	0.02602	2.58	0.3876	35.49	545.8	510.3	456.9	53.44	0.0695	1.0474	0.9779
64	114.7	0.02605	2.53	0.3953	36.65	546.0	509.3	455.8	53.45	0.0717	1.0458	0.9741
65	116.9	0.02608	2.49	0.4016	37.81	546.1	508.3	454.8	53.46	0.0739	1.0441	0.9702
66	119.1	0.02611	2.45	0.4082	38.98	546.2	507.2	453.7	53.47	0.0761	1.0424	0.9663
67	121.3	0.02615	2.41	0.4149	40.14	546.3	506.2	452.7	53.48	0.0783	1.0408	0.9625
68	123.5	0.02618	2.37	0.4218	41.31	546.4	505.1	451.6	53.49	0.0805	1.0392	0.9587
69	125.7	0.02622	2.33	0.4292	42.49	546.5	504.0	450.5	53.50	0.0827	1.0376	0.9549
70	127.9	0.02625	2.29	0.4367	43.66	546.6	502.9	449.4	53.50	0.0849	1.0359	0.9510
71	130.3	0.02629	2.25	0.4444	44.84	546.6	501.8	448.3	53.51	0.0871	1.0343	0.9472
72	132.6	0.02632	2.21	0.4525	46.02	546.7	500.7	447.2	53.51	0.0893	1.0327	0.9434
73	134.9	0.02636	2.17	0.4608	47.20	546.8	499.6	446.1	53.51	0.0915	1.0311	0.9396
74	137.3	0.02640	2.13	0.4695	48.37	546.9	498.5	445.0	53.51	0.0937	1.0294	0.9357
75	139.7	0.02643	2.10	0.4762	49.55	547.0	497.4	443.9	53.51	0.0959	1.0279	0.9320
76	142.1	0.02647	2.06	0.4854	50.73	547.0	486.3	442.8	53.50	0.0981	1.0262	0.9281
77	144.5	0.02651	2.027	0.4933	51.91	547.1	495.2	441.7	53.50	0.1003	1.0245	0.9242
78	146.9	0.02655	1.994	0.5015	53.09	547.1	494.0	440.5	53.49	0.1025	1.0229	0.9204
79	149.3	0.02659	1.961	0.5099	54.27	547.2	492.9	439.4	53.49	0.1046	1.0213	0.9167
80	151.8	0.02662	1.928	0.5187	55.46	547.2	491.7	438.2	53.48	0.1068	1.0197	0.9129
81	154.4	0.02666	1.896	0.5274	56.64	547.3	490.6	437.1	53.48	0.1089	1.0182	0.9093
82	157.0	0.02670	1.864	0.5365	57.82	547.3	489.5	436.0	53.47	0.1110	1.0166	0.9056
83	159.7	0.02674	1.833	0.5456	59.00	547.3	488.3	434.8	53.46	0.1132	1.0150	0.9018
84	162.5	0.02678	1.803	0.5546	60.18	547.3	487.1	433.6	53.45	0.1154	1.0134	0.8980
85	165.3	0.02682	1.774	0.5637	61.37	547.4	486.0	432.6	53.44	0.1175	1.0118	0.8943
86	168.1	0.02686	1.745	0.5731	62.55	547.5	484.9	431.5	53.43	0.1197	1.0102	0.8905
87	170.9	0.02690	1.717	0.5824	63.76	547.6	483.8	430.4	53.42	0.1219	1.0087	0.8868
88	173.8	0.02694	1.690	0.5917	64.97	547.6	482.6	429.2	53.41	0.1240	1.0071	0.8831
89	176.6	0.02698	1.663	0.6013	66.17	547.6	481.4	428.0	53.41	0.1262	1.0056	0.8794
90	179.5	0.02702	1.636	0.6112	67.38	547.6	480.2	426.8	53.40	0.1283	1.0040	0.8757
91	182.4	0.02706	1.610	0.6211	68.59	547.6	479.0	425.6	53.39	0.1305	1.0025	0.8720
92	185.3	0.02710	1.584	0.6313	69.80	547.7	477.9	424.5	53.37	0.1326	1.0009	0.8683
93	188.3	0.02714	1.559	0.6414	71.01	547.7	476.7	423.4	53.36	0.1348	0.9994	0.8646
94	191.3	0.02718	1.534	0.6519	72.21	547.7	475.5	422.2	53.35	0.1370	0.9979	0.8609
95	194.4	0.02722	1.509	0.6627	73.41	547.7	474.3	421.0	53.34	0.1392	0.9963	0.8571
96	197.5	0.02726	1.484	0.6739	74.64	547.7	473.1	419.8	53.32	0.1412	0.9948	0.8536
97	200.7	0.02730	1.460	0.6849	75.84	547.7	471.9	418.6	53.31	0.1434	0.9933	0.8499

TABLE 1—*Continued*

Temperature, Deg. Fahr.	Pressure, Lb. per Sq. In. Abs.	Specific Volume, of Liquid, Cu. Ft. per Lb.	Specific Volume of Vapor, Cu. Ft. per Lb.	Density, Lb. per Cu. Ft.	Heat of Liquid, B.T.U.	Total Heat, B.T.U.	Latent Heat B.T.U.			Entropy		
							Total	Internal	External	Liquid n	Vapor N	Difference $N-n$
98	203.9	0.02734	1.437	0.6959	77.07	547.7	470.6	417.3	53.29	0.1455	0.9917	0.8462
99	207.2	0.02739	1.415	0.7067	78.29	547.7	469.4	416.2	53.28	0.1477	0.9902	0.8425
100	210.5	0.02743	1.393	0.7179	79.51	547.7	468.1	414.9	53.26	0.1498	0.9887	0.8389
101	213.8	0.02747	1.371	0.7294	80.72	547.7	466.9	413.7	53.25	0.1519	0.9872	0.8353
102	217.2	0.02751	1.350	0.7407	81.95	547.7	465.7	412.5	53.23	0.1540	0.9857	0.8317
103	220.6	0.02755	1.329	0.7524	83.16	547.7	464.5	411.4	53.21	0.1562	0.9841	0.8279
104	224.0	0.02760	1.309	0.7639	84.37	547.7	463.3	410.2	53.19	0.1584	0.9826	0.8242
105	227.5	0.02764	1.290	0.7752	85.59	547.7	462.1	408.9	53.17	0.1606	0.9811	0.8205
106	230.8	0.02769	1.271	0.7868	86.81	547.7	460.8	407.6	53.15	0.1627	0.9796	0.8169
107	234.3	0.02773	1.252	0.7987	88.04	547.7	459.6	406.4	53.13	0.1649	0.9781	0.8132
108	238.0	0.02777	1.233	0.8110	89.26	547.6	458.3	405.2	53.11	0.1670	0.9766	0.8096
109	241.6	0.02782	1.214	0.8237	90.48	547.6	457.1	404.0	53.09	0.1691	0.9751	0.8060
110	245.3	0.02786	1.196	0.8361	91.70	547.6	455.9	402.8	53.05	0.1712	0.9736	0.8024
111	249.0	0.02790	1.178	0.8489	92.92	547.5	454.6	401.6	53.03	0.1733	0.9721	0.7988
112	252.8	0.02794	1.161	0.8613	94.15	547.5	453.3	400.3	53.01	0.1755	0.9706	0.7952
113	256.6	0.02799	1.144	0.8741	95.38	547.5	452.0	399.0	52.98	0.1776	0.9692	0.7916
114	260.4	0.02803	1.128	0.8865	96.61	547.4	450.7	397.7	52.95	0.1797	0.9677	0.7880
115	264.2	0.02807	1.111	0.9001	97.85	547.4	449.5	396.5	52.92	0.1818	0.9662	0.7844
116	268.1	0.02812	1.095	0.9132	99.10	547.3	448.2	395.3	52.89	0.1839	0.9747	0.7808
117	272.1	0.02817	1.078	0.9276	100.34	547.2	446.9	394.0	52.86	0.1860	0.9632	0.7772
118	276.2	0.02821	1.061	0.9425	101.58	547.2	445.6	392.8	52.83	0.1881	0.9617	0.7736
119	280.3	0.02825	1.046	0.9560	102.82	547.1	444.3	391.2	52.80	0.1902	0.9602	0.7700
120	284.5	0.02830	1.031	0.9699	103.06	547.0	442.9	390.1	52.76	0.1923	0.9587	0.7664
121	288.6	0.02834	1.016	0.9843	105.30	546.9	441.6	388.9	52.73	0.1944	0.9572	0.7628
122	292.7	0.02839	1.000	1.000	106.55	546.8	440.2	387.5	52.69	0.1965	0.9557	0.7592
123	296.9	0.02844	0.985	1.015	107.81	546.7	438.9	386.3	52.66	0.1986	0.9542	0.7556
124	301.2	0.02849	0.970	1.031	109.07	546.6	437.5	384.9	52.62	0.2007	0.9528	0.7521
125	305.5	0.02853	0.956	1.046	110.33	546.5	436.2	383.6	52.58	0.2028	0.9513	0.7485
126	309.9	0.02858	0.942	1.061	111.58	546.4	434.8	382.3	52.54	0.2049	0.9499	0.7450
127	314.3	0.02863	0.929	1.076	112.83	546.3	433.5	381.0	52.50	0.2070	0.9484	0.7414
128	318.8	0.02868	0.916	1.091	114.07	546.2	432.1	379.6	52.46	0.2091	0.9469	0.7378
129	323.3	0.02872	0.904	1.106	115.32	546.0	430.7	378.3	52.42	0.2112	0.9455	0.7343
130	327.9	0.02877	0.891	1.122	116.58	545.9	429.3	376.9	52.38	0.2133	0.9440	0.7307
131	332.5	0.02882	0.878	1.139	117.82	545.8	428.0	375.7	52.34	0.2154	0.9426	0.7272

DISCUSSION

A. J. Wood. We have suggested in the two papers on properties of ammonia, an addition of two new tables which do not agree one with the other, but, on the other hand, neither varies largely from accepted tables used in this country for the range of pressures common in practice. In view of this, these should be regarded as later valuable data rather than as finally accepted tables, which will at once set aside other tables. The investigations now under way may aid in the final agreement on a single table of recognized authority.

One of the most important properties to be dealt with is the latent heat of the saturated vapor. An examination of Fig. 3 of Mr. Mosher's paper shows that the results of the different tables for total latent heat vary most for temperatures above 100 deg. fahr. Barring values of Ledoux, the best agreement between the several authorities is for temperatures at which the ammonia is worked to advantage, that is below 100 deg. fahr. Substantial agreement among all is found at 70 deg. fahr. If in Fig. 3 are plotted values of the latent heat in the table based on Wobsa's second equation (see Marks' paper), it will be found that the results agree closely with the Dieterici values between 50 and 100 deg. fahr., and fall from 1 to 3 per cent below the proposed Goodenough values for this range. The differences in the new tables become less for the lower temperatures and in any event are not large. Cannot these differences be reconciled, bringing both tables to a substantial agreement?

The experimental value for the specific heat of liquid ammonia as found by the present writer and referred to in the tables given in the papers, was obtained by a method not followed by others and which may have certain merit, as it yielded check results with repeated trials. Briefly, it was as follows: A steel bomb holding nearly 59 grams of the liquid was constructed so as to insure its being completely filled before the tests began. Water of determined weight and temperature was allowed to drop at a given instant into a can partly filled with water, inserted in the center of a Mahler bomb calorimeter jacket. Into this can was placed another can containing the bomb, the water and ammonia being thoroughly agitated during the test. The radiation losses were carefully determined. Tests were carried to the lower temperature, but radiation losses for these temperatures

did not check in successive tests. The trials have not been repeated for the lower temperatures.

Wm. Earl Mosher replied that he realized that any tables based on the existing experimental data cannot be accepted as final, but only as representing our present knowledge of the subject and of a temporary nature. Final tables cannot be adopted until much more careful experimental work has been done. Such an investigation is now being carried on by the United States Bureau of Standards, but it will be a long time before the investigation can be completed and tables calculated from the results of the investigation.

In the meantime it is highly desirable for the sake of uniformity in calculations that some table be adopted as a temporary standard. Professor Marks has shown that the Wobsa table is superior to the Hybl table, and it is believed that the arguments presented in this paper are sufficient to demonstrate the superiority of the proposed table to the Wobsa table.[1]

As Professors Marks and Wood have said, the differences between the Wobsa table and the proposed table are very small for the range of pressures encountered in practice. The Wobsa table has been calculated, however, only with temperature as the argument and only at 9 deg. intervals; the new tables, on the other hand, have been prepared for every degree of temperature and for every pound of pressure throughout the working range. In addition, tables of the liquid and superheated vapor have been prepared and also a Mollier diagram. It would seem that the score of convenience would be sufficient to warrant the adoption, as a temporary standard, of the proposed tables.

[1] Further arguments concerning this point are included in Mr. Mosher's paper as it will appear in full as a bulletin of the Engineering Experiment Station of the University of Illinois.

No. 1365

EXPERIMENTS WITH NORTH DAKOTA LIGNITE IN A STEAM POWER PLANT AND A GAS PRODUCER

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Member of the Society

North Dakota has an area of approximately 70,000 square miles, of which about one-half is underlain with workable seams of lignite coal, varying from a few feet to 35 feet in thickness, which it is quite probable will some day exert a great influence upon its manufacturing and engineering industries. According to the State Geological Survey reports, the number of workable beds in southwestern North Dakota in 1908 were known to be 21. These are distributed through from 1000 to 1300 ft. of strata and range from 4 to 35 ft. in thickness. The aggregate thickness of the coal in these beds is 157½ ft.[1] Figs. 1 and 2 show the way in which the coal crops out[2].

2 It has been estimated by some that the North Dakota lignite deposits comprise one-sixth of the known coal deposits of the United States, but their extent and value are doubtless unappreciated, if indeed they are known, by the majority of the citizens of the State. Comparatively small amounts of the coal are mined annually, though the amount is increasing and will doubtless increase even more rapidly as the people learn to appreciate its value and to burn it in such a manner as to utilize more fully its heat units. It is regarded by many as an almost worthless fuel because it will not stand shipping in dry, warm weather without disintegrating and because it has been used in the ordinary furnace under boilers with poor results. It is used principally at points near the mines for power and domestic purposes and with

[1]N. D. State Geological Survey, 2d, 3d, and 5th biennial reports.

[2]N. D. State Geological Survey, 1908, Plates 13 and 15.

Presented at the Annual Meeting 1912, of The American Society of Mechanical Engineers

the present cost at the mine and freight rates and the present method of burning, it does not seem able to compete with the cheaper grades of Eastern bituminous coal at points a few hundred miles from the mines. Its use, however, is increasing, as shown by a statement received from the state engineer's office

FIG. 1 COAL BED OF THE GREAT BEND GROUP EXPOSED ON LITTLE MISSOURI NEAR THE HARMON RANCH. TOTAL THICKNESS OF COAL, 16 FT.

that 271,928 tons were mined in 1904, 385,882 tons in 1910, and 466,600 tons in 1911.

PRELIMINARY EXPERIMENTS

3 Before presenting and discussing the results of the experiments with North Dakota lignite as a fuel in a steam and producer gas power plant, the author will present the results of some rather simple but interesting preliminary experiments with the same coals as were used in the power plant tests. They show the nature of the fuel and its behavior when exposed to heat and

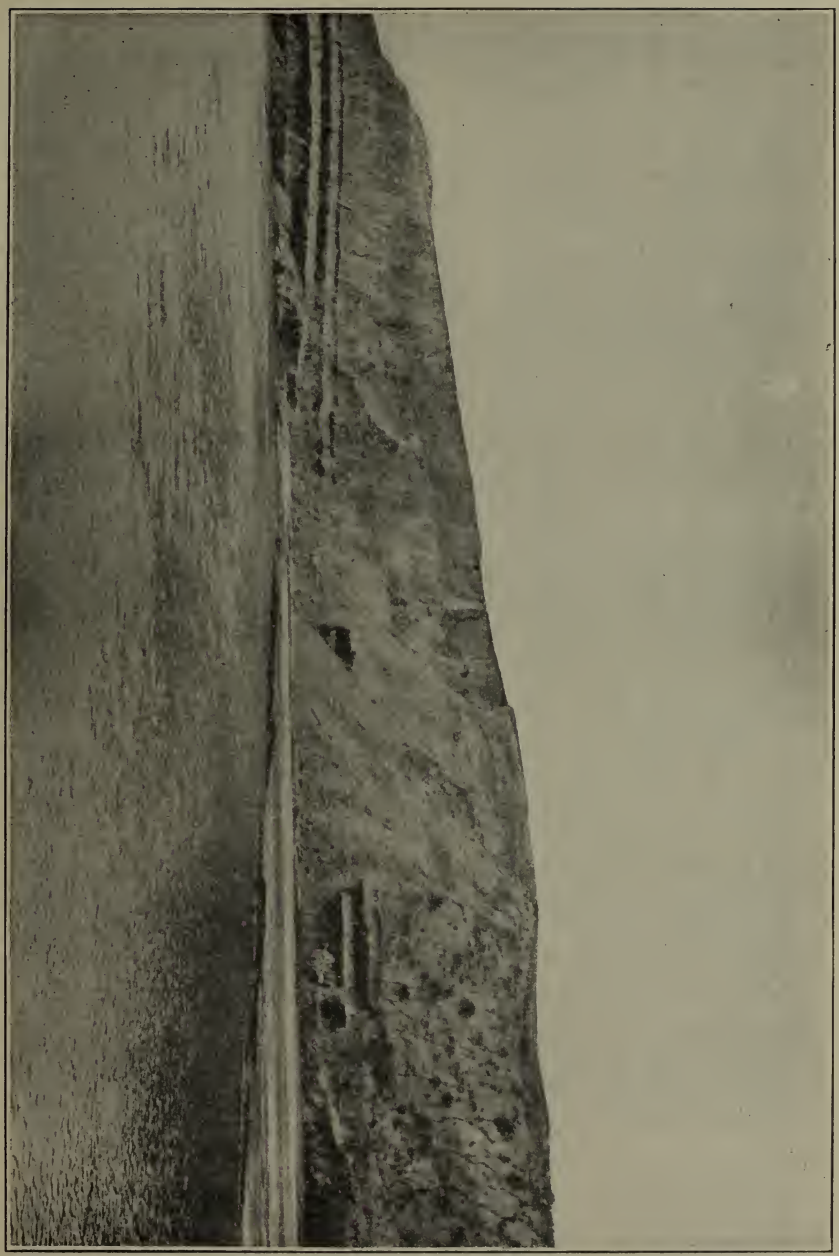

FIG. 2 COAL BEDS OF THE MEDORA GROUP EXPOSED ON LITTLE MISSOURI RIVER NEAR THE MOUTH OF ASH CREEK. TOTAL THICKNESS OF COAL, 16 FT. 5 IN.

to the drying action of the atmosphere and throw much light upon the results obtained in the power plant tests.

4 The coal used consisted of two car loads, one shipped in January 1911, from Williston, N. D., here designated as coal No. 1, and a second from Wilton, N. D., shipped in August 1911, referred to as coal No. 2.

5 Three samples of coal No. 1, weighing about 14 lb. each, were taken from different sections of the pile and weighed, submerged in water for 48 hours and again weighed. The gain in weight was but a fraction of 1 per cent, indicating that the coal was practically saturated with water when mined as it was shipped in cold weather and could not have given up or taken up much moisture in transit.

6 Coal No. 2 was shipped in August 1911, and was stored out of doors until needed for the tests. It was exposed to the elements during the summer and fall and naturally gave up some of its surface moisture, but the coal in the interior of the pile was quite wet in the winter when the tests were made. A block of this coal weighing 12 lb. 10 oz., which was taken from a point 2 ft. beneath the surface of the pile, submerged in water for 48 hours and again weighed, gained 14 oz., or 6.9 per cent in weight. The author was not present when coal No. 2 was received, hence, samples were not taken at that time.

7 To show the readiness with which the lignite gives up its moisture, four samples weighing respectively 30, 38½, 74 and 83 lb. were taken from coal No. 2 and placed in a room on March 16, 1912, and weighed from time to time. During the first week, the temperature was about 60 deg. fahr. and during the remainder of the time about 70 deg. On March 18, the coal had given up 2.43 per cent of its weight, on March 20, 3.76 per cent, on March 21, 6.09 per cent, on March 27, 9.2 per cent, April 1, 11.31 per cent and on April 16, 12.63 per cent. Figs. 3, 4, 5 and 6 were taken of these four samples on March 18, 20, 27 and April 16 and show the way in which the coal disintegrated as it dried. Fig. 7 shows the results after these four samples had been dropped 8 ft. upon a concrete floor, and indicates that when the coal is dry it is very friable.

8 To show how the coal disintegrates when stored for some time under cover, on April 6, 1912, a sample of 100 lb. was taken from a bin inside the building which had been filled with coal

No. 2 since August 1911. The bin contained about 4 tons. A small proportion of the coal (perhaps one-sixth) was in large chunks ready to fall into pieces with a light blow. The coal had been pulverized somewhat by firemen in shoveling it from the bin. However, neglecting the large chunks, the sample was a fair average of the coal. This sample was screened successively on sieves with ⅛ in., ¼ in. and ½ in. mesh; that which remained on the finer sieve was screened on the next coarser. Of the coal 26½ per cent passed through the screen with the ⅛ in. mesh, 24½ per cent passed through the screen with the ¼ in. mesh, 18 per cent passed through the screen with the ½ in. mesh and 31 per cent remained on the last. All would have passed through a 2½ in. ring.

9 Numerous proximate analyses were made, the results of which showed the green coal to contain approximately 35 per cent of moisture, and the dry coal to contain approximately 43 per cent of volatile matter, 50 per cent of carbon and from 7 to 9 per cent of ash. Difficulty was experienced in determining the percentage of moisture. It was finally decided to call that moisture which was driven off at a temperature below 240 deg. fahr. It was thought that some of the volatile matter was driven off at temperatures slightly above that. It was readily seen that gases were driven off at comparatively low temperatures as shown by the analytical balance and by the odors given off.

10 North Dakota lignite is classed as one of the youngest coals. In appearance it is black when freshly mined, but as it dries, it becomes lighter in color and is often a dark brown. Before it has become sufficiently dry to disintegrate, it somewhat resembles charcoal in that it frequently shows very clearly the grain of the wood and knots of the trees from which the coal has been formed; but upon drying it cracks both with and at right angles to the grain as seen in Figs. 3-7. The line of cleavage in breaking is along the grain. Occasionally a piece will be found which has lost its grain and at fracture resembles anthracite in appearance, having a luster, but this disappears as the moisture is driven off.

11 From these preliminary experiments, it will be seen that this fuel may contain (*a*) large quantities of moisture, the two coals analyzed containing over 35 per cent of moisture, which is readily given up, causing the coal to crack and disintegrate; (*b*)

FIG. 3 COAL AFTER BEING 2 DAYS INSIDE OF BUILDING

FIG. 4 COAL AFTER BEING 4 DAYS INSIDE OF BUILDING

FIG. 5 COAL AFTER BEING 11 DAYS INSIDE OF BUILDING

large proportions of volatile matter, some of which is driven off at comparatively low temperature; and (*c*) a small percentage of ash. These characteristics are very important and must be taken into consideration when designing furnaces and gas producers to utilize it. In the furnace it disintegrates and if burned

FIG. 6 COAL AFTER BEING 31 DAYS INSIDE OF BUILDING

FIG. 7 SAMPLES OF COAL AFTER FALLING 8 FT. UPON A CONCRETE FLOOR

on grates suitable for a coking coal, it will, when heated to red, run through the grate like so much coarse sand, especially if it be disturbed with a slice bar. Thus it will be seen that it should be burned on a grate with small openings and with a strong draft, for when it disintegrates it settles or packs down, restricting the air passages. One would also expect, since it contains a large proportion of volatile matter, that fire-brick arches or dutch ovens would be desirable in burning it under boilers.

12 An appreciation of these important properties and characteristics shown by the preliminary experiments will help in understanding the difficulties experienced in conducting the evaporative and gas producer tests and interpreting the results.

EVAPORATIVE TESTS

13 Evaporative tests, using North Dakota lignite under the boilers, were made in the mechanical laboratory of the College of Mechanical and Electrical Engineering at the State University of North Dakota. This plant consisted of two 70-h.p. horizontal fire tubular boilers, one of which was equipped with a Detroit automatic stoker and a dutch oven 7 ft. long projecting out in front of the boiler, while the other was provided with an ordinary furnace and rocking grates which were suitable for burning Eastern bituminous coals. After several preliminary experiments, this last-mentioned furnace was modified by placing fire-brick arches or tiles over three-fourths of the grate and by substituting perforated plates for the rocking grates, the perforations being ½ in. in diameter and 35 per cent of the area of the grate.

14 The plant was provided with natural draft which was adequate when burning Eastern bituminous coal to drive the boilers much beyond their rated capacities, but which proved inadequate when burning lignite. Several tests were conducted, varying from four to ten hours in duration. While they were unsatisfactory in that the boilers could not be driven at their rated capacities, they indicated conclusively that a much stronger draft would be necessary to keep the furnaces at a white heat and to force the boilers to their rated capacities. A small fan was available for producing a forced draft. It was not large enough to supply the necessary pressure but served its purpose for, with it, the furnace could be kept at a white heat, which was impossible without it. It was not thought advisable, however, to drive the fan at the necessary high speed for any considerable time and the results obtained with natural draft had to suffice. The author, wishing to confirm the results secured in 1911 by the students, conducted three tests in the spring of 1912, with about the same experience and results. Funds were not available at the time for the installation of an induced draft so that elaborate tests were not undertaken, the experiments being valuable chiefly because of the experience and knowledge gained. It was clearly

demonstrated that with a dutch oven and a mechanical stoker with proper draft conditions, the coal can be economically utilized in a steam power plant.

15 Table 1 gives the more important economic results obtained from these preliminary evaporative tests. For comparison the average of the results of 15 tests made by the U. S. Bureau of Mines, with this coal in a specially constructed furnace has also been included.[1]

TABLE 1 EVAPORATIVE TESTS

Furnace	Grate Area, Sq. Ft.	Duration of Tests, Hours	Per Cent Moisture in Coal	Dry Coal burned per Sq. Ft. of Grate Surface per Hour	Equivalent Evaporation from and at 212 Deg. per Lb. of Coal as fired	Equivalent Evaporation from and at 212 Deg. per Lb. Dry Coal	H.P. in Per Cent of Builders' Rating
Detroit stoker and dutch oven.....	20.25	10	35.77	12.20	4.22	6.58	67.4
Perforated plate grates with arches over 3/4 of the grate....	16	12	35.77	14.58	3.638	5.66	54.7
Special furnace used by Bureau of Mines.....	....	....	43.27	25.18	3.49	6.09	73.9 to 103.3

16 It will be noted that although the boilers were forced as hard as possible in the tests conducted by the author, they developed only from 54.7 to 67 per cent of the builders' rated capacity. In burning the coal on the chain grates under water tube boilers in the university power plant which had natural draft, a similar experience resulted. The experiment was begun with the furnace at a white heat, but as the lignite was fed into the furnace the fire-brick arches and tile on the water tubes cooled to a much lower temperature and the boiler could not be driven at anything like its normal rating.

17 An induced draft system was recently installed in the university power plant, and an induced draft apparatus has been ordered for the experimental plant in the mechanical laboratory

[1] Bulletin No. 2, Bureau of Mines, pp. 18, 19.

which will be capable of giving almost any amount of draft that may be desired in connection with a series of tests which will doubtless be conducted in the near future.

GAS PRODUCER TESTS

18 Experiments with North Dakota lignite in the gas producer were conducted in the same laboratory, the object being to determine the feasibility of using this fuel in a producer gas power plant.

APPARATUS

19 The experimental plant consisted of a 50-h.p. Smith suction down-draft gas producer and a 55-h.p. Alberger tandem producer gas engine belted to a dynamo. The scrubber was driven by an independent motor. In the tests of 1911, the jacket and scrubbing water was supplied from the city mains but for the 1912 tests a motor-driven triplex pump was installed so that the cost in power of supplying this water could be determined. In the 1911 tests, the dynamo used to load the engine was a 25-kw. direct-current machine, but for the subsequent tests a 45-kw. direct-current three-wire generator with compensating coils was installed so as to be able to load the engine to its maximum capacity. Fig. 8 shows a sectional view of the gas producer and Fig. 9 a sectional view of the engine.

METHOD OF LOADING THE ENGINE

20 The engine was loaded by means of a dynamo which supplied current to the university light and power circuit and also to a water rheostat which could be used when there was not sufficient demand otherwise for power to utilize all that was produced. The dynamo was so connected that current could be taken from the main line to drive the generator as a motor in starting up the engine. This made a simple and very satisfactory way of starting the engine which, when started, would speed up, increasing the generated electromotive force, and cause the motor to become a generator supplying current to the line, the load carried by the dynamo being governed by the voltage which was controlled by a hand-regulated field rheostat.

21 The generator used during the tests Nos. 1-8 inclusive was an old 25-kw. direct-current generator formerly used in the university lighting plant; in tests Nos. 9-18 inclusive a new Fort

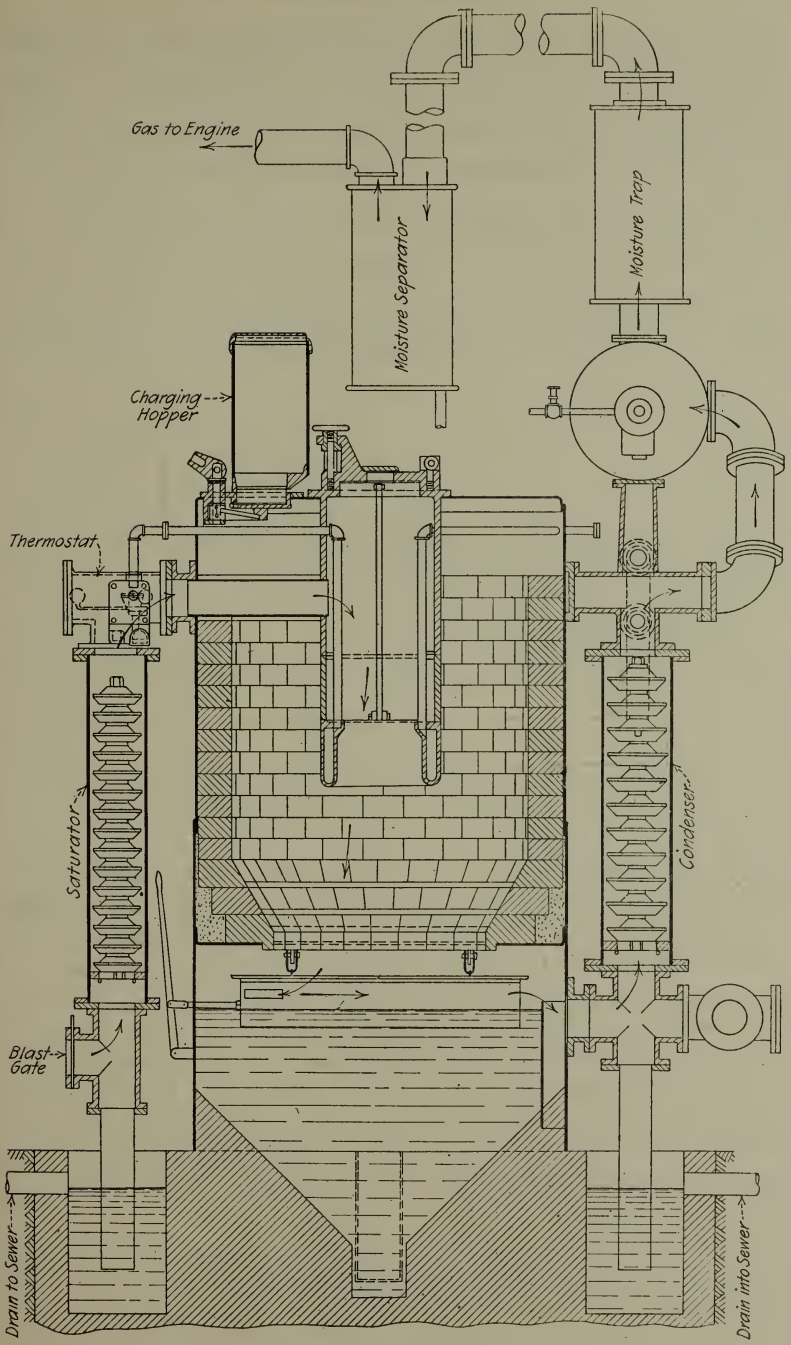

FIG. 8 SECTIONAL VIEW OF GAS PRODUCER

Wayne machine. This machine started up nicely but before test No. 12 had been finished, the commutator had become warped or out of true, causing much sparking at the brushes. This was remedied, however, before test No. 13 was run by turning down the commutator, after which it gave no trouble.

METHOD OF DETERMINING THE POWER GENERATED, AND COAL CONSUMED

22 The power delivered to the switchboard by the generator was measured, and the coal consumed per kw-hr. at the switch-

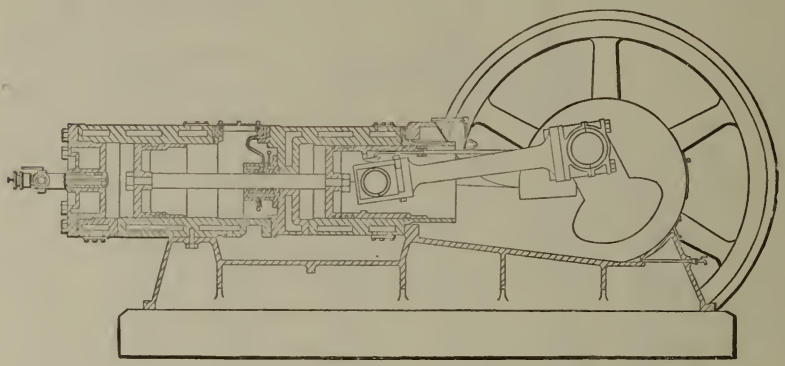

FIG. 9 LONGITUDINAL SECTION OF GAS ENGINE

board determined. It was the original intention to determine the efficiency of the generator so as to get the total power developed by the engine, but time did not permit before the armature of the generator was needed to replace a burned-out armature in a motor in the university power plant, so that the windage, friction and internal losses of the generator used in 1911 could not be obtained. However, since the machine was loaded nearly to its rated capacity most of the time, it was assumed in determining the brake horsepower of the engine that the generator had an efficiency of 90 per cent and that the belt transmission gave an efficiency of 98 per cent, both of which are probably higher than their true value.

23 As no means was at hand for measuring or determining the amount of water used by the engine jackets or the gas scrubber in the tests of 1911, accurate records of the power used by the

scrubber were not taken. To determine the net power available for outside purposes, it would be necessary to charge against such a plant all of the power consumed by the auxiliary apparatus. Fortunately funds were available in the fall of 1911 so that a motor-driven triplex pump was installed to supply the jacket and scrub water, also a more efficient motor was installed for driving the scrubber. In determining the net power available for outside purposes in the tests of 1911, it was assumed that the power consumed by the auxiliaries was the same as during the tests of 1912 when apparatus was available.

24 In determining the power developed by the engine in the tests of 1912, it was assumed that the belt transmission gave an efficiency of 98 per cent. The losses in the generator were determined by experiments in which the windage, friction, iron and copper losses were carefully determined and curves plotted showing the efficiency of the generator at various loads. The low efficiency of the generator may doubtless be explained by the fact that the machine was developing but little more than 50 per cent of its rated capacity.

25 In determining the coal used during the test, care was taken that the producer was full when the test began and when it ended. The coal charged in the meantime was considered as the coal used during the test. Before starting up the blower for the next test, the producer was poked to make sure there were no large voids and filled as before. The coal charged at this time was called the " standover " losses. A similar method was followed after fire had been " blown up " and the amount needed to fill the producer just before starting the test was called the " starting-up " losses.

COAL AND GAS ANALYSES

26 At the beginning of the first series of tests, Nos. 1-8, Table 2, the apparatus for analyzing the gases was broken so that but one analysis of the gas of these eight tests was made. Proximate analyses of the coal used in each test were made to determine the moisture. In the second and third sets of experiments, or those conducted in 1912, a complete analysis of the gases was made. The incomplete determinations of CO_2, O and CO were made with the Orsat flue gas apparatus. It is interesting to note that the composition of the two coals was almost identically the same.

TABLE 2 GAS PRODUCER PLANT POWER TESTS

No. of Test	Duration, Hr.	Coal Used	Moisture in Coal, Per Cent	Dry Coal Used	Moisture in Coal, Per Cent	Dry Coal Used	Coal as Fired per Hour	Dry Coal Used per Hour	Kw. Delivered at Switchboard	Kw. Consumed by Scrubber and Pump	Kw. at Switchboard Available for Outside Purposes	Power Consumed by Generator, H.P.	Brake H.P. of Engine Assuming an Efficiency for the Belt Transmission of 98 Per Cent	Coal as Fired per Kw-Hr. at Switchboard
1	8	632	32.9	424.1	32.9	424.1	79.0	53.0	14.18	3.71	10.47	21.10	21.53	5.57
2	8	1017	33.1	680.4	33.1	680.4	127.1	85.0	21.21	3.71	17.50	31.57	32.21	5.99
3	7¼	875	35.9	560.9	35.9	560.9	120.6	77.2	20.86	3.71	17.15	31.04	31.67	5.77
4	27½	2575	16.0	2163.0	16.0	2163.0	93.6	78.6	22.64	3.71	18.93	33.70	34.38	4.13
5	4	425	35.8	272.9	35.8	272.9	106.2	68.2	21.24	3.71	17.53	31.62	32.26	5.00
6	8	850	38.0	527.0	38.0	527.0	106.2	65.8	21.27	3.71	17.56	31.66	32.30	4.99
7	16	1700	36.5	1079.5	36.5	1079.5	106.2	67.4	22.67	3.71	18.96	33.75	34.43	4.68
8	6¼	500	37.2	314.0	37.2	314.0	76.9	48.3	18.5	3.71	14.79	27.54	28.10	4.10
9	24	2960	20.4	2356.0	20.4	2356.0	122.3	98.1	18.3	3.71	14.59	28.71	29.21	6.68
10	9½	1255	20.4	999.0	20.4	999.0	131.5	105.1	19.36	3.71	15.65	30.16	30.78	6.78
11	11½	1365	20.4	1086.0	20.4	1086.0	118.7	94.1	19.87	3.71	16.16	30.88	31.49	5.98
12	12	1125	20.4	895.0	20.4	895.0	93.7	74.5	19.12	3.71	15.41	29.82	30.43	4.90
13	9¾	1205	35.77	774.0	35.77	774.0	123.6	79.38	22.78	3.71	19.07	34.94	35.66	5.42
14	5 1/3	1072	33.0	718.0	33.0	718.0	201.1	134.7	21.31	3.71	17.60	32.91	33.58	9.43
15	9	1478	23.35	1132.0	23.35	1132.0	164.2	125.7	22.99	3.71	19.28	35.03	35.74	7.14
16	11	1351	23.35	1035.0	23.35	1035.0	122.8	94.0	22.88	3.71	19.17	34.92	35.64	5.36
17	7 2/3	820	23.35	628.0	23.35	628.0	107.0	82.0	24.54	3.71	20.83	37.37	38.13	4.40
18	24	3013	23.35	2309.0	23.35	2309.0	125.0	96.2	23.18	3.71	19.47	35.45	36.17	5.39

TABLE 2—Continued

No. of Test	Coal as Fired per Kw-Hr. Available for Outside Purposes	Coal as Fired per Brake H.P.-Hr.	Dry Coal per B.H.P.-Hr.	CO_2, Per Cent	O, Per Cent	CO, Per Cent	H, Per Cent	CH_4, Per Cent	N, Per Cent	Number of Samples Analyzed	B.t.u. per Cu. Ft.	Suction in Inches of Water at			
												Engine Throttle	Scrubber	In Pit below Grate	Air Inlet
1	7.56	3.67	2.46	....	....	....	....	....	....	....	....	....	....	....	....
2	7.26	3.94	2.64	....	....	....	....	....	....	....	....	....	....	....	....
3	7.02	3.81	2.43	....	....	....	....	....	....	....	....	....	....	....	....
4	4.94	2.72	2.28	5.10	4.65	17.58	9.31	2.3	61.06	....	111.4	....	....	....	....
5	6.05	3.29	2.11	....	....	....	....	....	....	....	....	....	....	....	....
6	6.05	3.28	2.03	....	....	....	....	....	....	....	....	....	....	....	....
7	5.61	3.08	1.95	....	....	....	....	....	....	....	....	....	....	....	....
8	5.20	2.74	1.72	....	....	....	....	....	....	....	....	....	....	....	....
9	8.38	4.22	3.35	11.9	2.4	11.6	....	....	....	2	....	15.8	13.6	4.9	0.25
10	8.40	4.27	3.41	9.2	2.5	16.7	....	....	....	2	....	20.7	15.2	7.3	0.7
11	7.34	3.77	3.00	10.8	2.3	14.0	....	....	....	3	....	21.9	17.4	7.6	0.1
12	6.09	3.08	2.44	10.0	2.6	12.5	....	....	....	4	....	21.1	16.8	10.7	0.18
13	6.48	3.46	2.22	9.8	1.0	10.8	....	....	....	3	....	23.0	20.0	6.0	0.2
14	11.42	5.99	4.01	....	....	....	....	....	....	....	....	23.6	18.0	4.61	0.7
15	8.52	4.59	3.51	....	....	....	....	....	....	....	....	23.4	19.1	4.4	0.5
16	6.40	3.44	2.63	9.13	1.64	14.11	16.53	0.88	57.7	12	109.85	18.2	16.0	7.5	1.27
17	5.13	2.8	2.15	6.08	3.8	10.62	18.97	4.2	56.25	4	152.7	26.9	20.8	4.7	0.07
18	6.45	3.47	2.65	8.59	3.7	11.7	8.04	4.77	63.18	9	115.42	28.8	21.7	7.5	0.33

Tests Nos. 1–8 were made between February 2 and May 24, 1911; tests Nos. 9–18 between February 6 and May 10, 1912. In all of these a Smith suction down-draft 50-h.p. gas producer and an Alberger tandem 55-h.p. gas engine were used; while in Nos. 1–8 a Westinghouse 25-kw. and in Nos. 9–18 a Fort Wayne 45-kw. generator was employed. The lignite coal in Nos. 1–8 was from the J. Bruegger mine, Williston, N.D., and in Nos. 9–18 from the Washburn Mine, Wilton, N. D. An analysis of samples taken from piles out of doors showed for coal No. 1, volatile matter, 44.2 %; fired carbon, 50.1 %; ash, 5.7 %; moisture, 35.9 %; for coal No. 2, volatile matter, 43.58 %; fixed carbon, 50.33 %; ash, 6.09 %; moisture, 35.77 %.

DESCRIPTION OF TESTS

27 Test Nos. 1-8 inclusive were conducted at intervals of from one to two weeks apart. After a test had been run, the producer was filled and enough coal brought into the building during the following week to run the next test. This enabled much of the coal to thaw out, though frequently coal was charged into the producer in the frozen state. The coal used in test No. 4, however, had been in the boiler room for some weeks which caused it to be much drier than that used in the other seven tests, but the author is inclined to believe that the 16 per cent moisture recorded is much too small. It is possibly due to poor sampling or to the loss of moisture between date of sampling and date of analyzing.

28 These tests were conducted with the idea of getting the maximum power possible out of the engine, but the capacity of the generator limited this, while tests Nos. 8-12 were planned to determine the efficiency of a plant, such as an electric light plant where the load would be variable and the plant operating from but 6 to 12 hours per day, standing idle the remainder of the 24 hours. For this reason, little effort was made to keep the load uniform. Ammeter and voltmeter readings were taken at half-hour intervals, but the power consumed during these four tests was measured by means of a carefully calibrated watt-hour meter. The power developed during tests Nos. 1-8 was determined from the ammeter and voltmeter readings which were taken every 15 minutes; during tests Nos. 13-18 inclusive, the power was likewise determined but the readings were taken at intervals of 20 minutes each, the loads in tests Nos. 1-8 and in 13-18 being kept as nearly uniform and as large as seemed feasible.

TROUBLES EXPERIENCED

29 It was observed that at the engine throttle, in the scrubber suction and in the pit beneath the grate of the producer, the suction increased as the tests progressed, apparently due to fine ash and clinkers closing up the air passages. This would restrict the gas at the engine causing its speed to reduce, which in turn would lower the voltage and cause the load to be thrown off automatically.

30 On one occasion the suction became so great that air

was pulled down through the ash conveyor into the bottom of the producer, with a resulting explosion that shook the fire up violently, blowing much fine ash out through the top of the central tube, which happened to be open. The engine automatically threw off its load, the dynamo operating as a motor driving the engine until the spent gases had been exhausted, when the engine immediately speeded up and took on its normal load. The suction at the different parts of the plant was greatly reduced after this explosion so that it operated satisfactorily during the remainder of the test.

31 Another source of trouble was that the strong suction would pull ash and fine particles of coal over into the bottom of

FIG. 10 CLINKERS TAKEN FROM GAS PRODUCER

condensing chamber which gradually became clogged and finally shut off the supply of gas with the natural results.

32 It is possible that with more experience the producer might be operated to fuse practically all of the ash into large clinkers, which could be removed through the top of the central tube of the producer. A large part of the ash was thus fused but evidently enough escaped to cause trouble. Fig. 10 shows some of the clinkers removed.

33 The disintegrating action of the coal when subjected to a high temperature caused the coal to settle down and the bed to become dense with restricted air passages, which also tended to increase the suction. Violent shaking of the grate seemed to help reduce this high suction and occasionally the suction would drop to but a few inches when large clinkers were removed.

34 It was observed in conducting the preliminary tests of

1912 when running tests on successive days using fuel with a large percentage of moisture, that the engine would satisfactorily carry its load during the first five or six hours and then the fire would become chilled and the gas so poor that the engine would scarcely run with no load. It was also observed when this happened that the fire in the producer had burned unevenly, being hotter on one side than on the other, doubtless allowing air to pass down through the bed without burning the coal. This action can perhaps be explained by the fact that the coal which had stood in the producer over night had become sufficiently dry to burn satisfactorily and that the trouble began when the fresh coal charged during the test reached the fire zone, chilling the fire because of its large amount of moisture and causing it practically to die out in certain sections and to become black at the lower end of the central tube where the fire was usually bright when operating under normal conditions.

35 The trouble is one that can be easily overcome either by drying the coal before shipping from the mine or by storing it for a few weeks in a dry, warm place where it will readily give up a large part of its moisture.

REMARKS

36 The starting and standover losses of the producer plant were surprisingly small. From a number of records made, it was estimated that the starting-up losses did not exceed 50 lb. even when the producer had been standing idle for one or two weeks and required 35 to 45 minutes to get good gas. The starting-up losses when the producer was operated on successive days were much less as good gas would be delivered at the engine in from 10 to 20 minutes from the time the blower was started. The engine would often carry full load within 15 or 20 minutes after starting the blower.

37 The standover losses for fourteen different observations covering 867 hours amounted to 3043.5 lb. or an average of 3.3 lb. per hour. The periods of standover varied from 12 to 300 hours, the standover losses for the short periods being more per hour than for the long periods. Taking 777 hours of the longer standover periods, the average per hour of the standover losses was but 1.92 lb. Comparing the starting-up and standover losses of the producer gas plant with those of a steam

plant, the author believes the former has a decided advantage over the latter.

38 It should also be borne in mind that a commercial plant would doubtless be much larger than this experimental plant and would give correspondingly better results. Furthermore, it is quite possible that the power consumed in scrubbing the gas was much larger than was necessary, as it was feared trouble would arise from tar getting into the engine.

39 Doubtless a larger amount of scrub water was used than was necessary, thus greatly increasing the amount of power consumed in scrubbing the gas. The amount of power consumed by the scrubber could be doubled or trebled simply by increasing the quantity of scrub water used. No trouble was experienced, however, from tar getting into the engine cylinders. The cylinder head of the engine was removed for inspection from time to time. A small amount of gummy substance adhered to the counterbore of the cylinder but the cylinder barrel was well lubricated. It was thought that the substance adhering to the counterbore was largely cylinder oil and carbon partially burned, but it may have been partially burned tar. In either event, there was not enough to cause the piston rings to stick or leak or to interfere with the action of the admission and exhaust valves.

40 A lengthy discussion comparing the economic results of the coal consumed per brake horsepower or per kw-hr. at the switchboard with the amounts required by steam plants of similar size is not necessary. It should be apparent to all that the cost of power in terms of pounds of lignite per b.h.p. or per kw-hr. at switchboard is much less than it would have been in a steam power plant of like size.

41 None of those interested in these tests was familiar with the operation of gas producers so that all had much to learn about the care and manipulation of the plant. Doubtless much better results would have been secured by those experienced in the handling of this type of plant. The author, however, feels gratified with the results for it should be expected that with more experience in handling the producer a much better economy would be obtained.

ACKNOWLEDGMENT

The author wishes to acknowledge his indebtedness to Dean E. J. Babcock of the School of Mines; to Mr. J. Brennan, instructor and fellow in the School of Mines; to Messrs. N. W. Bennington, F. H. Bradshaw, Jr.,

R. A. Heising, A. W. Kishpaugh, G. C. Budge and G. T. Challoner, seniors in the College of Mechanical and Electrical Engineering, for the results of the numerous tests, made by them in preparation of theses; to Mr. P. E. Henwood, instructor in the college; and also to the Bruegger Coal Company, Williston, N.D., and the Washburn Lignite Coal Company, Wilton, N.D., for their donation of a car load of coal.

DISCUSSION

A. M. Levin. These experiments are interesting and instructive in so far as they deal with the nature of fuel in general, and with its evaporative qualities; but in so far as they deal with lignite producer gas they are not in any way fair tests of the fuel for producer use. The author himself doubts that they are; and his caution that doubtless under more favorable conditions better results could have been obtained, may serve to offset to some extent the unfavorable impression the extraordinary test figures are bound to give.

One fact which the evaporative tests bring out is that under the boiler the lignites are only the poorest kind of fuel, for one reason, if for no other, that they call for something like twice the boiler capacity any ordinary steam coal would require. Another fact about the lignites which the experiments fail to bring out is that as producer fuel they are ideal. Some lignites have proved to be and others, that as yet have not, should be given a fair trial. The chief object with the gas producer is, or should be, the handling of poorer grades of fuel among which the lignites must be counted.

Of the 18 producer tests of the Grand Forks' series there is but one that shows a reasonable fuel consumption per unit output. Mr. Crouch generously puts the blame for the shortcoming of the tests on a presumed inexperience of the experimenters; yet it seems entirely more likely that the blame should be laid to the employment of a wrong type of gas generator. There is not the slightest doubt but that the down-draft principle is wrong with respect to lignite fuels. These fuels, as the experiments bring out, disintegrate when heated in the fuel bed, so that by the time they reach the hearth of the generator they are very fine. When, therefore, in the down-draft producer the fine fuel becomes intermixed with the ashes which form in the greatest quantity at the top of the fuel bed, it should naturally be expected that the bed would become somewhat compact. Consider for a minute that all

the fine loose particles in the fuel bed act in the way of so many small valves that tend to seat themselves over the interstices between the grains of fuel and it must be evident that to draw the gases downward in the direction in which these valves close is wrong. Instead, the draft should be upward, in which direction these valves can lift.

Unfortunately, the amount of tar separated in the cleaning of the gas is not recorded. This item would have been interesting for the sake of comparison with what can be accomplished by the up-draft producer in the way of fixing the tarry products of the fuel. However, when some tar will have to be removed from the lignite gas whatever the type of the producer, it would not pay just for the sake of producing a somewhat smaller percentage, to ruin the gas process entirely by adhering to the down-draft principle. An attempt to run the producer so hot as to fuse all the ashes into clinkers would normally result in a leaner gas and a loss of sensible heat in the extremely hot gases. As ordinarily arranged, the power gas producer should be run consistently cool and with a slow current through the fuel bed. There is a time element involved in the breaking up of tarry hydrocarbons as well as in the reduction of the CO_2 gas.

In order to gain a better idea of what can be obtained in the way of returns by the use of lignite in the power gas producer it may be of interest to compare Tests Nos. 17 and 18 of the Grand Forks' series with one of the Bureau of Mines experiments at the Norfolk experimental station on a North Dakota lignite of identically the same quality as the fuels used in these tests; and also with a test on a Washington lignite, which the writer had occasion to make some years ago at Olympia, Wash. This test was fully described in Power[1] and it may be of interest because of the fact that although the fuel was quite high in ash, it proved by the use of a singularly suitable producer to give a high efficiency.

Test No. 17 of the Grand Forks' series is chosen because it is the best of the entire series, and Test No. 18 because it is rather poor, though not the poorest. The proximate analyses of the fuels used in the tests referred to were as given in Table 3, and the results obtained as in Table 4.

In the Olympia and Norfolk test figures allowance has been made for steam supplied to the producers.

[1] December 27, 1910.

TABLE 3 PROXIMATE ANALYSES OF FUELS USED IN TESTS

	Fuel	Moisture	Volatile Matter	Fixed Carbon	Ash	Heat Value per lb., B.t.u.
Grand Forks, Tests Nos. 17 and 18....	N. D. lignite	23.35	33.9	38.4	4.35	8400*
Norfolk Test No. 2.....	N. D. lignite	39.56	27.78	26.3	6.36	6802
Olymphia.....	Washington lignite	16.2	28.90	32.1	22.8	7560

* The fuel value of this lignite is not given, but it was evidently quite high; an estimate gives about 8400 B.t.u.

TABLE 4 RESULTS OBTAINED IN UTILIZING FUELS IN POWER GAS PRODUCER

	Gas Analysis						B.t.u. per Cu. Ft.	Economy Coal as Fired per B.t.u-Hr.	Gas Pressure, Inches of Water		Tar Per Cent Weight of Fuel
	CO	CH_4	H	O	CO_2	N			Leaving Producer	Entering Engine	
Grand Forks, Test No. 17.....	10.62	4.2	18.97	3.8	6.08	56.25	152.7	2.8	−4.7	−26.9	....
Grand Forks, Test No. 18.....	11.70	4.77	8.04	3.7	8.59	63.18	115.4	3.47	−7.5	−28.8	....
Norfolk, Test No. 2.....	20.90	4.85	14.33	0.23	8.69	51.00	188.5	2.23	2.1	0	3
Olymphia.....	17.2	4.70	24.70	0.6	11.2	41.6	184	1.75	−1.75	− 2	7/8

The excessively high suction on the gas system of the Grand Forks' plant clearly indicates the real cause of the poor performance of the plant. Of the various losses accountable to this difficulty the following are of importance:

Direct loss due to increased resistance on the engine piston.

Loss due to inefficient combustion and reaction because of excessively high current velocity through the fuelbed, resulting in a lean gas.

Loss due to a reduced engine efficiency because of lack of density of the lean gas in the cylinder.

Loss because of a highly increased power requirement in cleaning the rarified gas.

The excessive suction caused evidently also a considerable leakage of air into the gas system. This, however, may only apparently have affected the heating value of the gas.

The Washington fuel gave a perfectly clean ash, entirely free from coal or clinkers, but the North Dakota fuel used in the Norfolk test is reported to have clinkered some. The cause may have been that the producer, in order to maintain a full load on the engine, was run somewhat above its capacity. Naturally a lean coal, as lignite, will require a somewhat greater producer capacity than a richer fuel.

THE AUTHOR. I appreciate the generous remarks of Mr. Levin, but I firmly believe that a more experienced person would have secured better economic results in the producer gas tests. It is not claimed in the paper that the suction down-draft producer is the type of gas generator best suited to the utilization of North Dakota lignite. Before installing this experimental unit in 1910, I secured information and advice from a large number of gas producer manufacturers, and found but very few who made producers in capacities of 50 h.p. or under, which they would guarantee to handle lignite satisfactorily. There was no question about large producers of other types handling this fuel, but I wanted a plant approaching in size that which might be installed in a small town or one large enough to demonstrate that the coal could be used in a small isolated plant of the producer gas type. The state of North Dakota has numerous small but rapidly growing towns or villages which are, or soon will be, requiring electric lighting and pumping plants. The

problem with them is what kind of a plant should be installed; so that this was regarded as an appropriate problem for investigation in North Dakota. In installing this unit I acted upon the advice given by some of the most prominent gas producer manufacturers, one of whom was at the time experimenting with a lignite producer and has since, I understand, perfected one, but I am not aware that his company builds them in small sizes.

In connection with the statement that exceptional economic results are not shown, I would state that it was not expected to secure these, but instead to show that the coal could be utilized profitably in that size of producer, in which I believe I succeeded.

As stated in my paper a larger amount of scrub water was doubtless used than was necessary, with the result that the power consumed by the scrubber was greater than was necessary. Had this been reduced to a minimum, the coal consumption per kilowatt-hour available at the switchboard for outside purposes would have been materially reduced.

Furthermore, the refuse which passed through the grate was larger than required, due to unnecessary shaking. This refuse was all considered as waste, while as a matter of fact it should have been recharged into the producer and have been more completely burned. It was not thought wise to do this while conducting the tests, but in subsequent tests this refuse was refired with good results.

There was not enough tar to warrant its being saved. It collected in the bottom of the pit mixed with ash and fine particles of coal which were drawn into the condensing chamber and washed into the pit along with the tar, etc.

Concerning the statement of Mr. Levin relative to the lignite being a poor steaming coal requiring a large boiler, I do not think the tests reported indicate it, but rather that a boiler utilizing this fuel in its furnace needs a large amount of grate surface, a strong draft, and a special furnace adapted to the burning of a fuel rich in volatile matter.

I have not tried to prove that this fuel could be used in a producer gas plant of 250 h.p. or over, for the United States Government has done that, but that it could be used profitably in smaller sized plants such as might be installed in a small town or manufacturing plant, and in this I think I succeeded.

No. 1366

THE BALTIMORE SEWERAGE PUMP VALVE

By A. F. Nagle, Bethlehem, Pa.

Member of the Society

In December 1906, the Baltimore Sewerage Commission awarded to the Bethlehem Steel Company a contract for building three sewage pumping engines of substantially the same design as the modern standard city waterworks pump, namely, "a self-contained, vertical, triple-expansion crank and flywheel, three single-acting plungers, condensing engine; each pumping engine to have a capacity of 27,500,000 gal. in 24 hr., with an average total head of 72 ft. (7 ft. suction lift); and to have a capacity for 50 per cent greater speed for short periods." The only deviation from the waterworks design was in the pump valves. These, instead of being made of rubber discs, 3 in. or 4 in. in diameter, spring operated, were to be as follows:

2 "They shall be rectangular flap valves swinging on horizontal hinges, faced with rubber of the best quality for the purpose, and *weighted;* the valve seats shall be of hard, tough bronze, and *the planes of their faces shall be inclined.* (Italics are the author's.) The hinges of the valves shall be protected, by hooding or otherwise, from the direct flow of the sewage passing through the valves; suitable stops shall be provided to limit the opening of the valves at the extreme edges to 2½ in., and the clear waterway through either suction or discharge valve shall not be less than 145 per cent of the area of the plunger."

3 In the spring of 1907, when the drawings were well advanced, the author was appointed by the commission to be their mechanical engineer and inspector at the Bethlehem Steel Company's plant. It had been decided to make the normal speed of the engines of 5 ft. stroke and 20 r.p.m., which would be increased to 30 revolutions to comply with the contract capacity.

Presented at the Annual Meeting 1912, of The American Society of Mechanical Engineers.

This speed would necessitate a plunger 40¼ in. in diameter to give the required volume of water. For the valve decks there had been tentatively planned thirty-four 4 in. by 12 in. and six 4 in. by 9 in. valves.

4 The sewage pump valve is far from being of a standard design. The commission's engineers had given the subject much study and made many inquiries before writing the paragraph

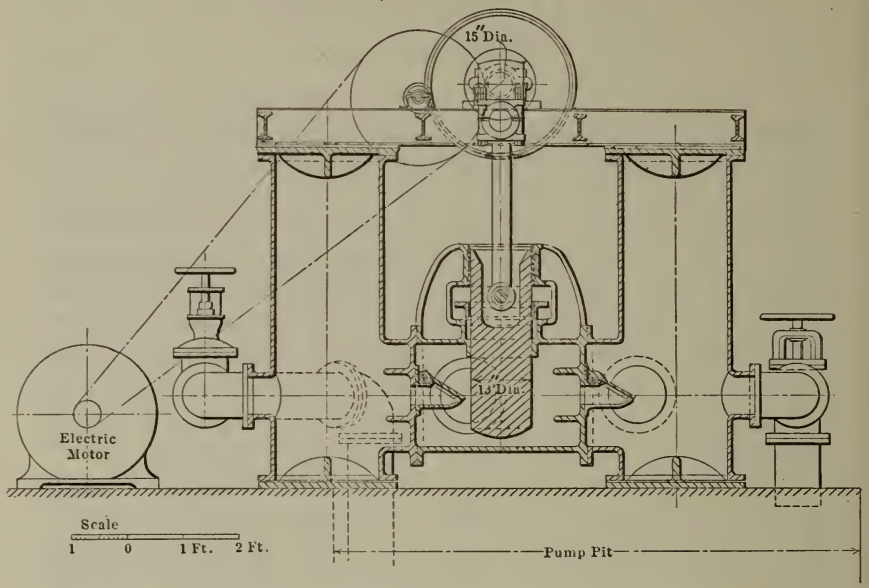

FIG. 1 PUMP FOR STUDYING VALVES DESIGNED FOR BALTIMORE SEWAGE

quoted, and yet it is noticeable that they refused to be specific on the essential elements of the valve, namely, *its seat-angle* and *its weight.* As there would be required over 700 of these bronze valves, it would be an expensive affair to make a change in them after completion. The writer therefore suggested to the management of the Bethlehem Steel Company that they construct a small pump for the purpose of trying out one set of the proposed valves. With this they were in hearty accord, and Fig. 1 shows a sectional elevation of the pump as made. It is hardly necessary to describe it, except to say that it is 13 in. in diameter by 15 in. stroke, making from 20 to 30 r.p.m.; being the exact equivalent of 6½ in. diameter by 5 ft. stroke, the proportional

ratio of plunger and valves. The 7-ft. suction lift was also reproduced.

5 In order to obtain definite knowledge of the action of the valve, there was attached to the arbor of its hinge a 5/8-in. shaft extending through the casing and a light stuffing box, and to this shaft were fastened suitable arms, levers and rods leading to an ordinary indicator from which the spring had been removed. In this way, a perfect trace of the lift of the valve could be obtained, examples of which accompany this paper. At the same time a pump card was taken and some of these are also reproduced.

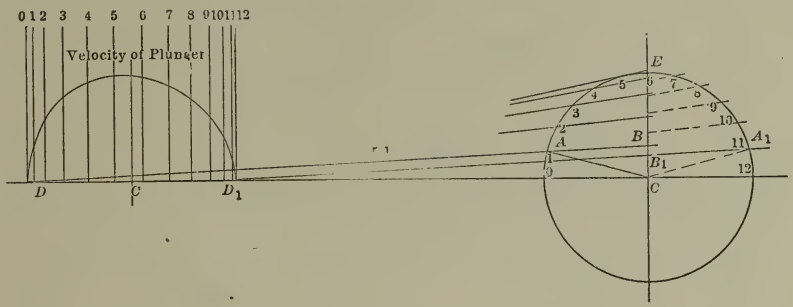

FIG. 2 GRAPHICAL METHOD FOR FINDING PLUNGER VELOCITY

6 The plunger velocity is dotted on the valve card, making the maximum plunger velocity agree with the highest lift of valve. The graphical method of finding this velocity is reproduced in Fig. 2, copied in substance from Rankine's Rules and Tables, page 236. The only difference in condition between this small pump and the main pump is that the relatively more rapid motion of the plunger in this case is in the lower end while in the main pump it will be at the upper end. The practical result of this arrangement is that the suction valve is favored in closing time, while in the main pump it will be the discharge valve that is thus favored.

7 It was decided to experiment with three different seat angles, 30, 45, and 60 deg. with the vertical plane, and weighted to different amounts. From the large number of tests made, two sets of cards for each of the three seat angles have been selected, at 20, 25, and 30 r.p.m.

8 The first test at a change of seat angle was always made with the normal or unweighted valve. Then followed one with

the suction valve loaded 5¼ lb., and the discharge valve 10.8 lb. There is no good reason, however, why the valves should not have been weighted alike.

9 Valve and pump cards are shown in Figs. 3, 4, 5, 6, 7, and 8, and data pertaining thereto are given in Table 1.

EXPLANATION OF TABLE 1

10 Columns 1 to 5 inclusive are self-explanatory. Columns

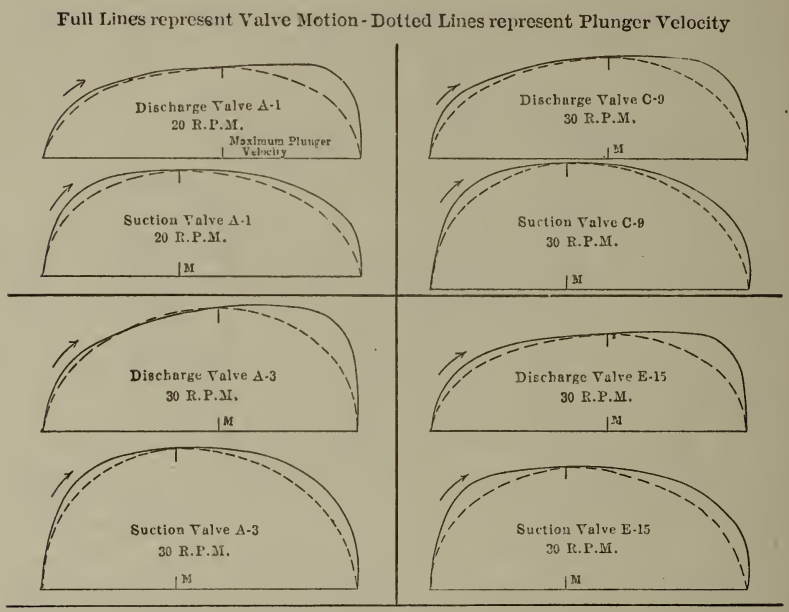

FIG. 3 VALVE CARDS, SEAT ANGLE 30 DEG.

6 and 7, weight of valve, is the entire weight before it is put in place. Columns 8, 9, 10, and 11, are obtained by a spring scale attached to the outer edge of the flap and noting the force required to start the valve; and again, the amount at its full lift of 2½ in., the scale being held, of course, at right angles with the flap. It should be noted that owing to friction of the hinge, it usually required from one to two pounds more to move the flap upward than downward, and the amount given in the table is the mean of the two.

11 Columns 12-13: The actual lift of the valve is obtained as follows: Owing to the intercession of certain cranks and levers,

the pump card trace is not exactly in line with the outer edge of the flap. To correct for this the valve cards as taken must be multiplied by the following factors: 1.25 for the 30 deg. seat; 1.25 for the 45 deg. seat; and 1.54 for the 60 deg. seat.

MECHANICAL ACTION OF THE VALVES

12 Mr. A. J. Mayer, on behalf of the Bethlehem Steel Com-

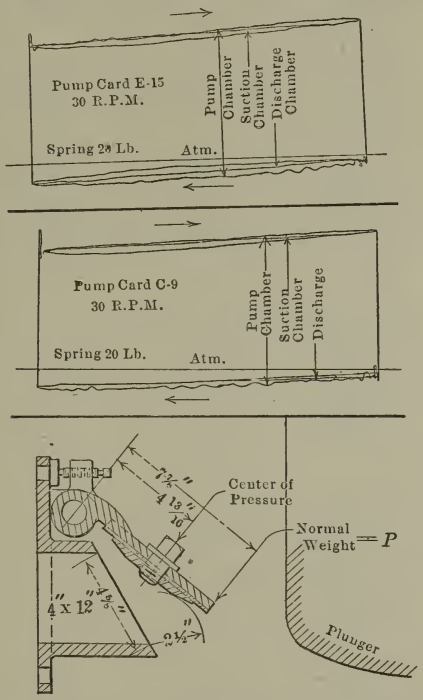

FIG. 4 PUMP CARDS, SEAT ANGLE 30 DEG.

pany, took all cards and made the following comments on the working of the pump valves:

Seat Angle, Deg.	Card No.	R.p.m.	Suction Valv	Discharge Valve
	A-1	20	Knocks slightly at closing. Very slight knock at opening.	Knocks at opening louder than suction. Very slight knock at closing.
30	A-2	25	Knocks slightly at closing. Very slight knock at opening.	Knocks at opening louder than suction. Very slight knock at closing.

Seat Angle, Deg.	Card No.	R.p.m.	Suction Valve	Discharge Valve
	A-3	30	Knocks slightly at closing. Very slight knock at opening.	Noises slightly increased at opening and closing
	C-7	20	Barely audible knock at closing. No noise at opening.	Pounds at opening. No noise at closing.
30	C-8	25	Barely audible knock at closing. No noise at opening.	Pounds at opening. Barely audible pound at closing.
	C-9	30	Barely audible knock at closing. No noise at closing.	Noise slightly increased at opening. Very slight pound at closing.
	21	20	No noise at opening or closing.	Knock slightly at opening and closing.
45	22	25	No noise at opening or closing.	Noises slightly increased.
	23	30	No noise at opening or closing.	Knocks loud at opening and closing.
	27	20	No noise at opening or closing.	Barely audible pound at opening. No noise at closing.
45	28	25	No noise at opening or closing.	Knocks slightly at opening and closing.
	29	30	No noise at opening or closing.	Knocks slightly at opening and closing.
	39	20	No noise at opening or closing.	Barely audible thud at opening. No noise at closing.
60	40	25	No noise at opening or closing.	Audible thud at opening, and closing.
	41	30	No noise at opening or closing.	Pounds slightly at opening. Audible thud at closing.
	45	20	No noise at opening or closing.	Barely audible thud at opening. No noise at closing.
60	46	25	No noise at opening. Slight thud at closing.	Makes thud at opening and closing.
	47	30	No noise at opening. Slight thud at closing.	Makes thud at opening and closing.

Seat Angle, Deg.	VALVE DIMENSIONS	Sq. In.
30	Area of Valve Seat Outside	82
	Area of Valve Seat Inside	55
	Ratio of Areas, 1 to 1.48	
	Area of Full 2½-in. Valve Lift	45
45	Area of Valve Seat Outside	101
	Area of Valve Seat Inside	72
	Ratio of Areas, 1 to 1.41	
	Area of Full 2½-in. Valve Lift	49.50
60	Area of Valve Seat Outside	107
	Area of Valve Seat Inside	80
	Ratio of Areas, 1 to 1.34	
	Area of Full 2½-in. Valve Lift	55

NOTE: It should be stated that owing to the nearness of the hub of the hinge to the body of the valve, only the front and two sides of the opening were taken as discharge area.

13 A point is now reached where there are facts to serve as a guide in the selection of a design for the valve, without any mathematical calculation whatever, the application of which is left to a later part of this paper. Referring for a moment to Mr. Mayer's remarks, attention should be called to the difficulty in distinguishing between a "thud" and a "knock" of a valve, either suction or discharge. The writer's opinion is that a "thud" always occurs at the *opening* of a valve, due to the difference between the inner and outer seat area, thus necessitating a greater pressure within to start to open the discharge valve than exists without; and the converse for the suction valve. All the pump cards show this greater or less pressure by a descent or an ascent of the indicator piston, which latter, however, falls to the normal pump pressure as soon as the valve opens. If there were air-pockets in a pump, the pressure of the air when released would send the valves violently against their stops. The soft rubber or leather facing allows this change of pressure to occur with nothing worse than a "thud."

14 In addition to the valve action, there is at this time the reversal of all stresses in the moving parts and journals, thus tending to increase the "thud" of the pump.

15 In contrast with a "*thud*" at *opening* is the "*knock*" at *closing*. This knock is due to the violent seating of the valve by the reversed action of the plunger, if, instead of coming gently

to its seat by its inherent weight as the plunger reaches the end of its stroke, it is not then seated.

SELECTION OF A DESIGN

16 A cursory examination of the valve traces or cards will show:

First, that *at opening*, every valve trace followed closely the theoretical or plunger-velocity line, except those

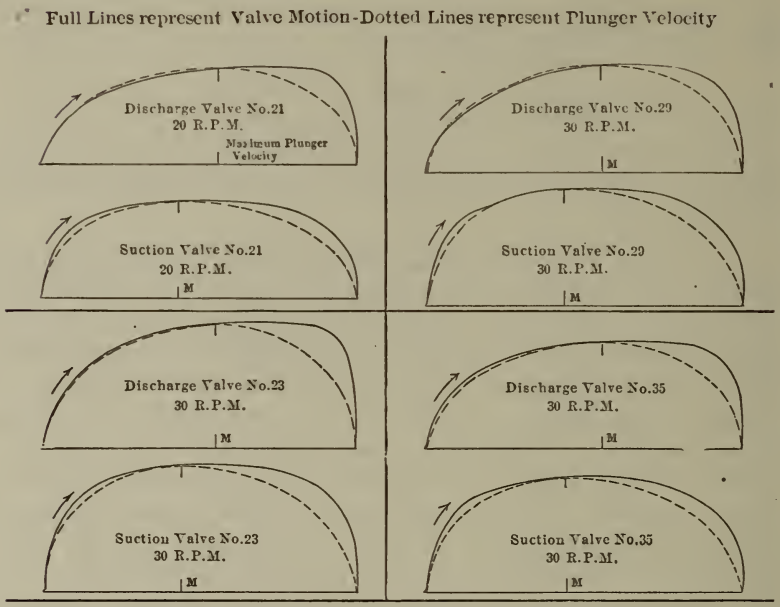

FIG. 5 VALVE CARDS, SEAT ANGLE 45 DEG.

of 30-deg. seat angle. In this case the valve acted like a swinging pendulum, in the impetus it received in opening, notwithstanding its natural increased weight, due to change of angle, carrying it beyond its proper position. In studying these cards, one must not lose sight of the different plunger velocities at the two ends of the stroke.

Second, in closing there was a constant improvement from the 30-deg. to 60-deg. seat angle; and, of course, with its increased weight. The heavier the valve the better it worked, mechanically, but the lift was reduced—

not desirable in a sewage valve—and the velocity through the valve increased.

17 In opening, every pump valve, either suction or discharge, if of constant or nearly constant weight, such as all gravity acting valves are, is *forced* by the plunger to follow its own varying velocities; but in *closing* there is no force or pressure to do this.

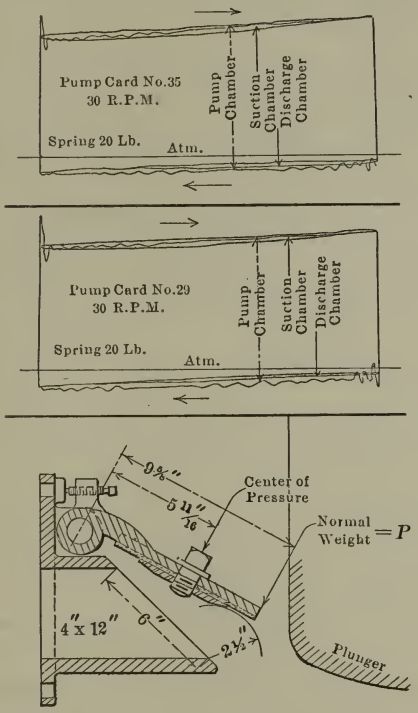

Fig. 6 Pump Cards, Seat Angle 45 Deg.

It has only its own inherent weight to move it, and its effective weight is diminished by several causes. First, there is the resistance of its hinge, probably very small, since the fit is very free; second, the water cannot fill in the back of a flat valve any more than it can flow into the space left by the square stern of a boat; hence the partial vacuum formed there increases the lifting force of the stream beneath it; and third, there is the inertia of the mass of the valve as well as of the water which has to change its position.

18 Perhaps the experience all have had in rowing a boat up or down a rapidly moving stream will give a good impression of the difference between closing and opening a pump valve.

19 At 30 revolutions the 60 deg. seat valve (unweighted) reached only 2 in. of its 2½ in. possible lift. To have made it light enough to go to its 2½-in. stop at this speed would have necessitated reducing its weight about one-third (see last line in

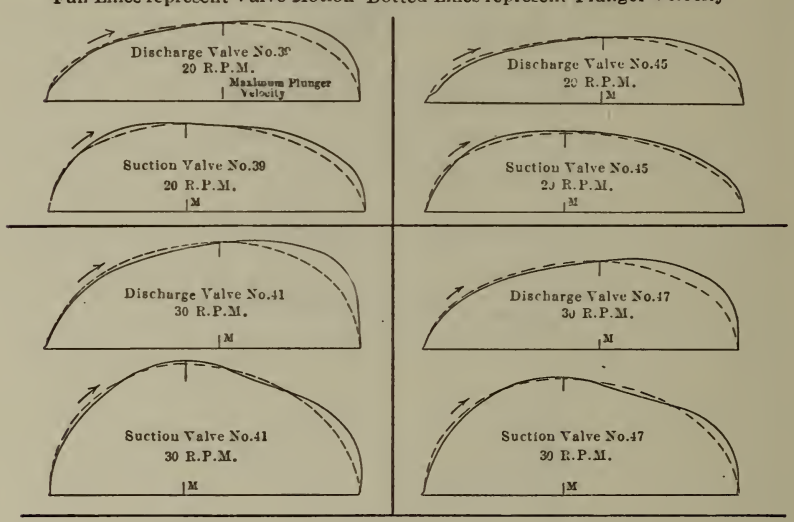

FIG. 7 VALVE CARDS, SEAT ANGLE 60 DEG.

Table 5) and with every indication that it would not reach its seat in closing in a way to prevent slamming. Valve cards 39-41 furnished the best proportions for a large lift and lowest velocity through the valve. Its weight of seat area per sq. in. was 0.221 lb. (See Table 3.) The extraordinary lift of 2½ in. of the suction valves 41 and 47 can be attributed to the proximity of the plunger at the lower end of its stroke, thus causing a back pressure at the valve outlet.

20 Fig. 9 is a reduced scale drawing of the valve as actually constructed. It weighs about 0.30 lb. per sq. in. of its seat area. It is of bronze throughout and has a renewable leather face. Its stop is cast on the body of the valve above it. There are 44 valves exactly alike, grouped into a 7-ft. steel valve deck. Figs. 11 and

12 are reduced photographs of these decks. Fig. 11 shows the valves closed and Fig. 12 the valves blocked open. Fig. 13 shows the pump end of the engine as it was erected in shop No. 2 of the Bethlehem Steel Company. The writer standing at the base, furnishes a scale to the pumps.

21 These pumps have now been in operation for several months and from the first turnover have worked so quietly one would hardly know there were any valves in them.

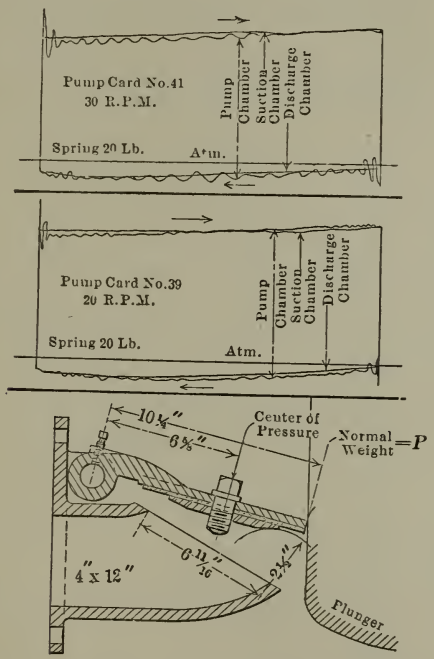

Fig. 8 Pump Cards, Seat Angle 60 Deg.

22 Aside from the design of the valve itself, great care was taken that all valves should be of like weight. When a group of valves are thus tied together, each is held in a very sensitive balanced position and any difference in weight is reflected in a different lift; hence a single light valve may be held back in closing and then slam to its seat. This was actually demonstrated by the action of the air-relief valve on top of the pump, which is a common swivel, swing check valve, without stop or definite weight.

23 Having described the valve as constructed, and conscious of its excellencies, shortcomings, cost of production and possible cost of renewals, the writer ventures to suggest that a cast-steel flap on a rounded seat would wear longer, cost much less to produce or repair and work fully as well, without a "thud," if the

TABLE 1 PUMP VALVE EXPERIMENTS. OBSERVED DATA

Seat Angle, Deg.	Card No.	R.P.M.	Pump Head, Ft.			Weight of Valve, Lb.		Normal Weight of Valve				Lift of Valve	
								Suction		Discharge			
			Suction	Discharge	Total	Suction	Discharge	Start	Top	Start	Top	Suction	Discharge
	1	2	3	4	5	6	7	8	9	10	11	12	13
30	*A*-1	20	7.59	67.1	74.7	27	28	8	11	8	10.5	1.60	1.38
30	*A*-2	25	7.71	65.8	73.5	27	28	8	11	8	10.5	1.85	1.63
30	*A*-3	30	7.93	66.9	74.8	27	28	8	11	8	10.5	2.15	1.90
30	*C*-7	20	7.25	66.9	74.2	32.2	38.8	11.5	14.5	14.0	17.5	1.46	1.14
30	*C*-8	25	7.71	65.6	73.3	32.2	38.8	11.5	14.5	14.0	17.5	1.65	1.29
30	*C*-9	30	7.93	65.0	72.9	32.2	38.8	11.5	14.5	14.0	17.5	1.92	1.53
45	21	20	6.35	65.9	72.2	30	31	10.1	11.1	10	11	1.50	1.50
45	22	25	6.35	66.7	73.0	30	31	10.1	11.1	10	11	1.70	1.69
45	23	30	6.35	64.6	70.9	30	31	10.1	11.1	10	11	1.96	1.94
45	27	20	7.03	67.1	74.2	35.2	41.8	13.7	15.2	16.7	19.5	1.36	1.24
45	28	25	7.03	67.9	74.9	35.2	41.8	13.7	15.2	16.7	19.5	1.50	1.38
45	29	30	7.03	66.8	73.8	35.2	41.8	13.7	15.2	16.7	19.5	1.80	1.64
60	39	20	6.80	67.3	74.1	34	32	13.2	14.2	12.7	13.6	1.69	1.48
60	40	25	6.80	66.1	72.9	34	32	13.2	14.2	12.7	13.6	2.20	1.81
60	41	30	6.80	66.1	72.9	34	32	13.2	14.2	12.7	13.6	2.50	2.00
60	45	20	6.80	64.5	71.3	39.2	42.8	17.6	19.1	21.1	22.4	1.54	1.25
60	46	25	6.80	65.6	72.4	39.2	42.8	.....	.....	.....	.....	1.92	1.51
60	47	30	7.36	66.1	73.4	39.2	42.8	.....	.....	.....	.....	2.26	1.68
			Aver..... 73.3										

seat were of bronze or possibly of cast iron. Half a dozen of these valves, as shown in Fig. 10, have been made for future trial.

AGREEMENTS

24 Having described the essential facts leading to the selection of the design of this sewage pump valve, the writer does not like to leave the subject without an effort to present a theory as a guide to the construction of similar valves. Aside from the evidence given by the valve cards of the better working of the 60-deg. seat valve than of the valves with 30 deg. or 45 deg. angles,

there is the important result of a much larger outlet area with the same lift of valve, and consequent lower velocity through the valve. This is shown in the following tabulation of results with the unloaded valves at 30 revolutions:

30 deg. seats 34.2 sq. in. outlet, 1.90 lift, 7.72 ft. velocity
45 deg. seats 38.6 sq. in. outlet, 1.95 lift, 6.48 ft. velocity
60 deg. seats 44.4 sq. in. outlet, 2.00 lift, 5.94 ft. velocity

TABLE 2 PUMP VALVE EXPERIMENTS. CORRECTIONS OF ORIGINAL DATA APPLIED TO DISCHARGE VALVE. SEE FIGS. 14 AND 15

Seat Angle, Deg.	Card No.	R.P.M.	Lift of Valve		Normal Weight of Valve			
			Log, In.	Corrected In.	Log		Corrected	
					Start	Top	Start	Top
	1	2	3	4	5	6	7	8
30	*A*-1	20	1.38	1.43	8	10.5	8	10.5
30	*A*-2	25	1.63	1.61	8	10.5	8	10.5
30	*A*-3	30	1.90	1.90	8	10.5	8	10.5
30	*C*-7	20	1.14	1.14	14	17.5	14	17.5
30	*C*-8	25	1.29	1.29	14	17.5	14	17.5
30	*C*-9	30	1.53	1.54	14	17.5	14	17.5
45	21	20	1.50	1.46	10	11	10	11.7
45	22	25	1.69	1.68	10	11	10	11.7
45	23	30	1.94	1.95	10	11	10	11.7
45	27	20	1.24	1.20	16.7	19.5	16.7	19.5
45	28	25	1.38	1.38	16.7	19.5	16.7	19.5
45	29	30	1.64	1.61	16.7	19.5	16.7	19.5
60	39	20	1.48	1.51	12.7	13.6	12.7	13.6
60	40	25	1.81	1.75	12.7	13.6	12.7	13.6
60	41	30	2.00	2.00	12.7	13.6	12.7	13.6
60	45	20	1.25	1.26	21.1	22.4	21.1	22.4
60	46	25	1.51	1.47	21.1	22.4	21.1	22.4
60	47	30	1.68	1.68	21.1	22.4	21.1	22.4

In the latter case, the throat area of 48 sq. in. is very closely approached. (See Fig. 17.)

25 The basis of all hydraulic calculations is the feet head, or pounds pressure, per square inch of area, and consequent velocity of flow in feet per second. This relation is fixed absolutely by natural law and if theoretical results are not obtained, one must look for some diverting cause, such as contraction, change of direction, or friction. These last effects must be ascertained by experiment. For simple forms such as pipes, the values are

well established; but for irregular shapes such as these valve openings present, no data exist, and from the limited amount of information gathered during the experiments, the writer is unable to make a complete analysis of the valves. They do give *something*, however, as a guide in future construction, which the writer will endeavor to present.

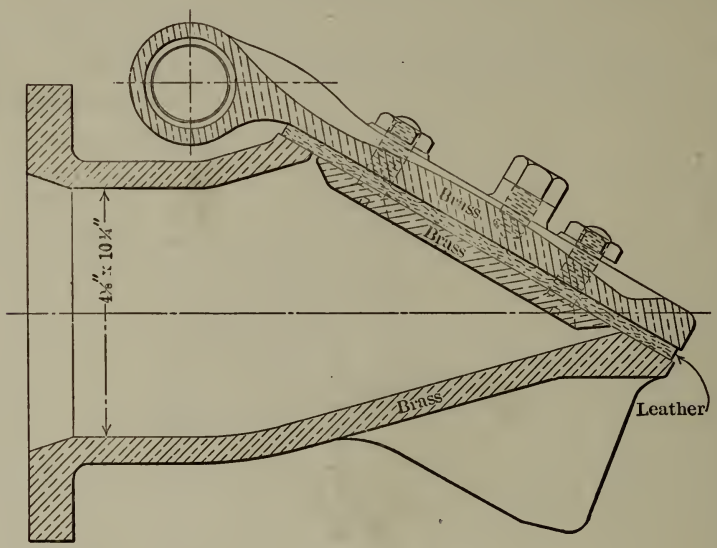

FIG. 9 VALVE AS CONSTRUCTED

26 The problems which it was attempted to solve are these:

First, knowing the weight—the normal weight—of a valve per square inch of seat area, how nearly will that weight, or pressure, give a velocity through the valve in agreement with the actual velocity found by experiment?

Second, knowing the lift of a valve at a given number of revolutions, how much will this lift be increased by an increase of revolutions?

Third, knowing the lift and weight of a valve at a given speed, what will be the lift if the weight be increased but the speed remain the same?

27 Before proceeding with the analysis it should be explained that it will be confined to the discharge valve only, and for these

reasons: In the first place, the writer has no theory of valve action that would distinguish between the two; second, certain discrepancies in the data, especially relating to the suction valve, led him to doubt their accuracy; and, third, there is no use in burdening the paper with more figures than are absolutely necessary.

28 As already explained, the original or log figures are given in Table 1. For the purpose of the calculations it has seemed

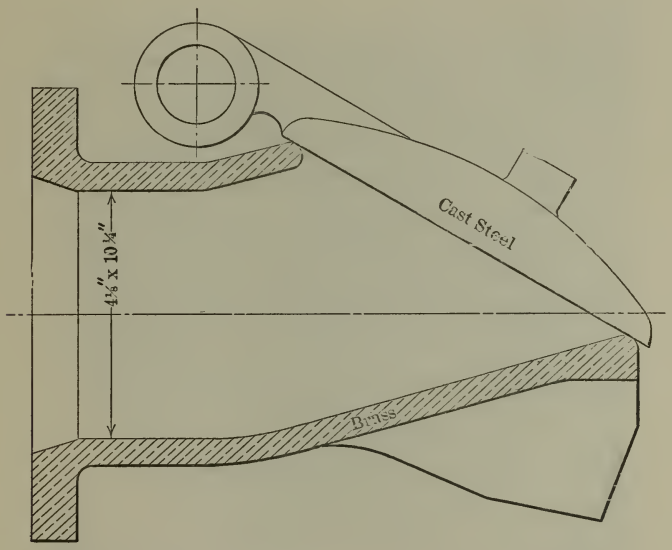

FIG. 10 DESIGN OF VALVE PROPOSED BY THE AUTHOR

necessary to make certain corrections upon these, as shown in Table 2 and in diagrams Figs. 14 and 15. Numerous small changes were made in the lift of the valve, but only one in the normal weight of the valve. These corrected values were used in all subsequent calculations, as exhibited in Table 3.

EXPLANATION OF TABLE 3

29 Column 14: The maximum plunger velocity is really slightly more than the crank velocity—for a 5 to 1 connecting rod it amounts to nearly 2 per cent, or 1.60 times the plunger travel, and this is given in this column. The main engine dimensions have been preserved rather than those of the experi-

mental pump, no change in calculated result being effected thereby.

30 Column 15: To facilitate subsequent calculations, there is given the plunger area corresponding to each valve (which is 33.00 sq. in.) multiplied by its velocity, giving the discharge per second in inch-feet.

31 Column 16: The area of valve opening is found by taking the full area opening given under "Valve Dimensions" in Par.

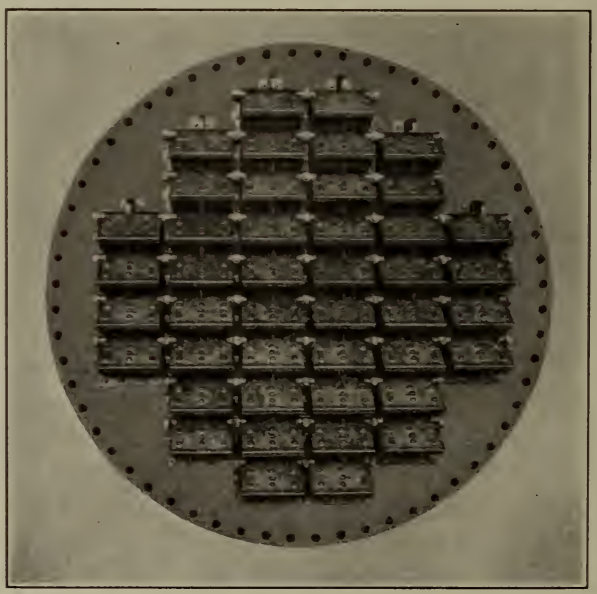

Fig. 11 View of Valves, Closed

12, for 2½-in. lift, and reducing it to its proportionate amount for the actual lift given in Table 2, column 4.

32 Column 17: The apparent velocity through the valve is an important result. It is found by dividing the rate of discharge of plunger given in column 15, by the area of valve opening given in column 16. Therefore, if any two adjoining values in columns 16 and 17 be multiplied the corresponding result given in column 15 will be obtained. This is called the *apparent velocity*, although commonly called the *actual velocity*, for it is what the velocity would be if the area given were wholly filled with the issuing stream; but there must have been some con-

traction, and hence a correspondingly higher velocity. The writer is unable to divide this apparent velocity into the various measured causes that produce it, and with this explanation rests this part of the discussion because he is unable to carry the analysis any further with the data at hand.

33 Column 18: Hydraulic engineers always desire to express the relation of *head* to *velocity*. Hence, if in column 17 the velocity through the valve is given, what is its corresponding

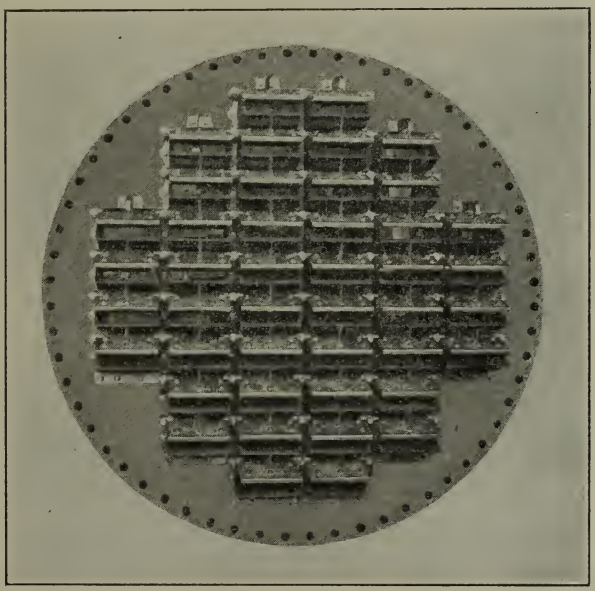

FIG. 12 VIEW OF VALVES, OPEN

head? Reducing the standard formula for feet head to pounds pressure, he has used the formula

$$\text{Vel.} = 12.20\sqrt{P} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

P being the pressure in pounds per square inch.

34 Column 19: The normal weight of valve is given in Table 2, only at its two extreme positions. It is necessary to know what it is at the position of its lift. If, for example, it takes 8 lb. to lift valve A-1 at the start, and 10.5 lb. at its 2½-in. lift, the weight increases just one pound for each inch of lift. Hence, if the lift is 1.43 in. the weight at 1.43 in. lift will be 9.43 lb. These values are found in this column.

35 Column 20: The normal weight of valve per square inch of seat area is found as follows: In the previous column (19) is given the total normal weight at *the edge of the valve.* Taking the center of pressure of the issuing stream at the center of the

TABLE 3 PUMP VALVE EXPERIMENTS. CALCULATIONS BASED UPON OBSERVED DATA APPLIED TO DISCHARGE VALVE ONLY

Seat Angle, Deg.	Card No.	R.P.M.	Max. Plunger Velocity, Ft. per Sec.	Rate of Max. Pump Discharge, In-Ft. per Sec.	Area of Valve Opening, Sq. In.	Apparent Velocity through Valve, Ft. per Sec.	Velocity Head due to Velocity through Valve, Lb.	Normal Weight of Valve at Lift, Lb.	Normal Weight of Valve per Sq. In. of Seat Area, Lb.	Velocity due to Normal Weight of Valve, per Sq. In. of Seat Area, Lb.	Ratio of Velocities of Col. 21 to Col. 17
	1	2	14	15	16	17	18	19	20	21	22
30	*A*-1	20	5.33	176	25.7	6.85	0.315	9.43	0.247	6.08	1.13
30	*A*-2	25	6.66	220	29.0	7.59	0.387	9.61	0.252	6.12	1.24
30	*A*-3	30	8.00	264	34.2	7.72	0.400	9.90	0.260	6.22	1.24
30	*C*-7	20	5.33	176	20.5	8.58	0.494	15.6	0.409	7.81	1.11
30	*C*-8	25	6.66	220	23.2	9.48	0.603	15 8	0.415	7.87	1.21
30	*C*-9	30	8.00	264	27.7	9.53	0.610	16.2	0.425	7.95	1.20
45	21	20	5.33	176	28.9	6.09	0.249	11.0	0.222	5.75	1.06
45	22	25	6.66	220	33.7	6.53	0.286	11.2	0.224	5.77	1.13
45	23	30	8.00	264	38.6	6.48	0.282	11.3	0.229	5.84	1 11
45	27	20	5.33	176	23.8	7 39	0.366	18.0	0.363	7.35	1.01
45	28	25	6.66	220	27.0	8.15	0.436	18.2	0.367	7.39	1.10
45	29	30	8.00	264	31.9	8.28	0.460	18.5	0.373	7.45	1.11
60	39	20	5.33	176	33.2	5.30	0.189	13.2	0.219	5.71	0.93
60	40	25	6.66	220	38.5	5.71	0.219	13.3	0.221	5.74	1.00
60	41	30	8.00	264	44.4	5.94	0.237	13.4	0.223	5.76	1.03
60	45	20	5.33	176	27.7	6.35	0 270	21.8	0.363	7.35	0.86
60	46	25	6.66	220	32.3	6.81	0.311	21.9	0.364	7.36	0.90
60	47	30	8.00	264	37.0	7.14	0.342	22.0	0.366	7.38	0.94

seat opening (which is probably not exactly correct, but which is the best that can now be affirmed), it is needed but to reduce the extreme total weight to this center, divide by the seat area in square inches and correct for loss of weight in water.

36 For the 30-deg. seat are these values, P being the total normal weight at lift, found in column 19:

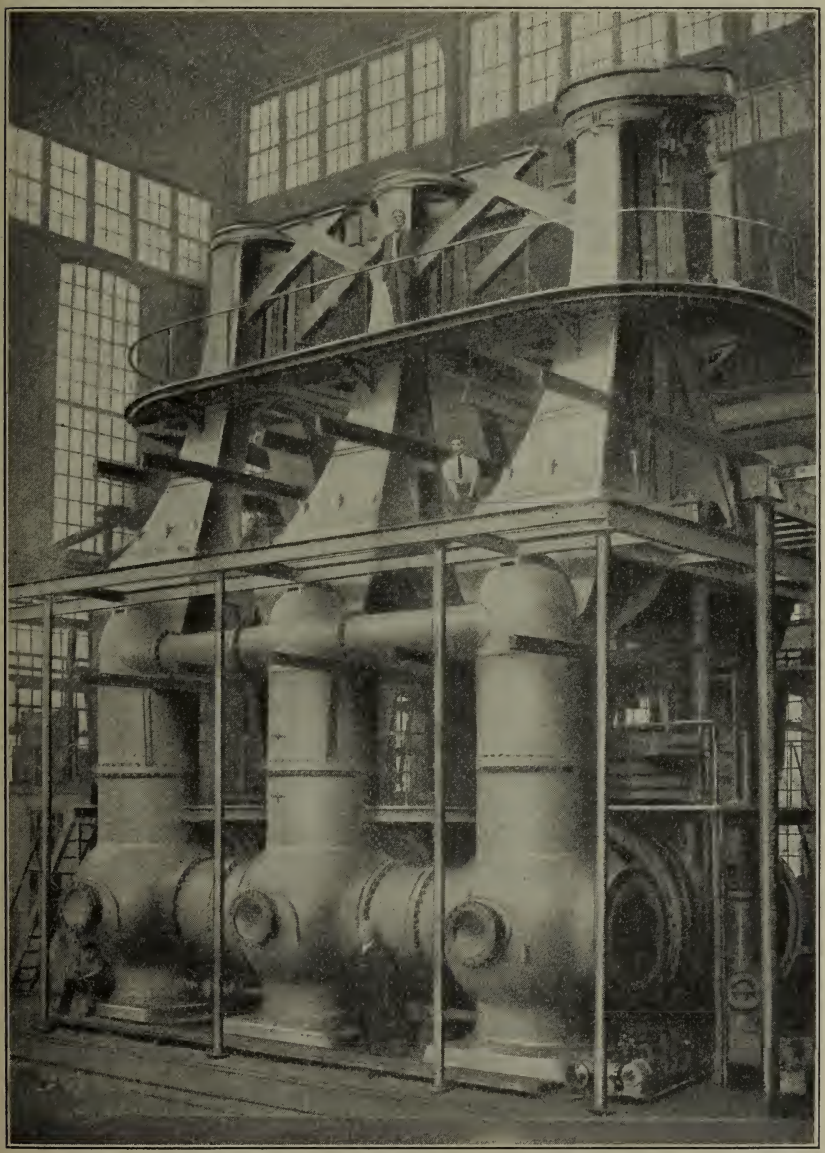

FIG. 13 PUMPS OF THE BALTIMORE SEWAGE PUMPING ENGINE AS ERECTED IN THE SHOP

$$W = \frac{7.87 \times 0.86 \times P}{4.81 \times 0.55} = 0.0256\ P \ldots\ldots\ldots\ldots [2]$$

For the 45-deg. seat these values become

$$W = \frac{9.62 \times 0.86 \times P}{5.69 \times 0.72} = 0.0202\ P \ldots\ldots\ldots\ldots [3]$$

For the 60-deg. seat

$$W = \frac{10.2 \times 0.86 \times P}{6.62 \times 0.80} = 0.0167\ P \ldots\ldots\ldots\ldots [4]$$

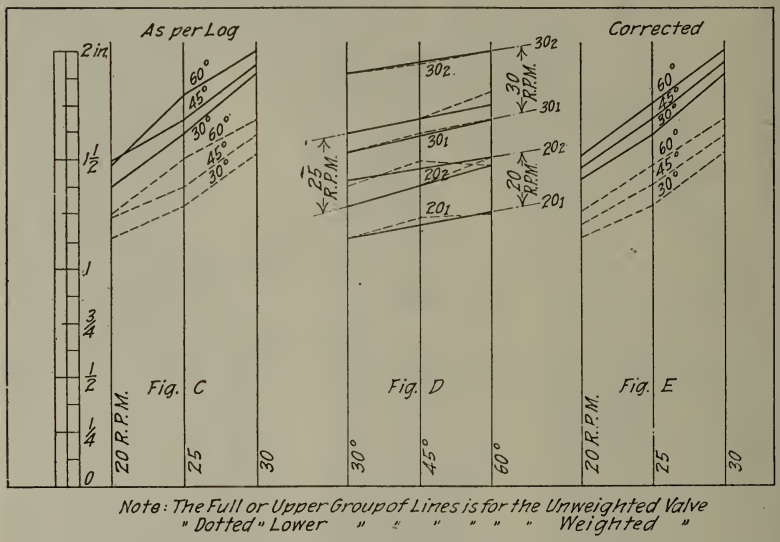

Fig. 14 Lift of Discharge Valve, corrected. See Table 2

37 Column 21: *Velocity* due to weight of valve in water per square inch of seat area, as given in column 20, is found by formula [1].

38 Column 22: It now remains to see how closely these calculated velocities agree with the apparent velocities as given in column 17. These ratios are found in this column, and for better study they are plotted in Fig. 16.

39 A check on the ratios of velocities in column 22 can be had by finding a similar ratio of the weight of the valve and weight or "head" due to the velocity through the valve, as given in columns 18 and 20, bearing in mind, however, that the velocities vary with the square root of the pressure or "head." For

example—taking the first line in Table 3, columns 18-20, the weights are respectively as 0.315 to 0.247, and their square roots as 0.561 to 0.500, or 1.12 to 1, practically the same as found in column 22, these slight divisions being due to the number of decimals carried in the calculations.

COMPARISONS

40 The point has now been reached where the results of these experiments can be considered with some degree of intelligence.

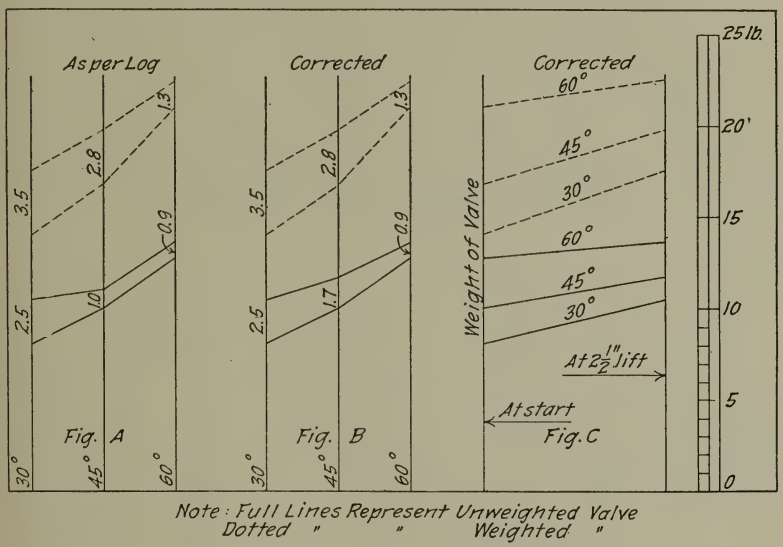

FIG. 15 TOTAL NORMAL WEIGHT OF VALVE AT START AND AT 2½-IN. LIFT. SEE TABLE 2

Turning to the first proposition propounded, how nearly does the calculated velocity, based upon the method here described, agree with the actual velocity through the valve as found by these experiments, on the whole, it seems to do so very closely, indeed, in the case of the 60-deg. seat and light valve, almost exactly. Rather than attempt to give mathematical expression to these deviations and apparent discrepancies, the designing engineer will use his own judgment as to the proper allowance to be made for them. Sufficiently accurate information has been obtained to prevent future mistakes in the construction of this type of valve as applied to the described conditions.

41 Second: In what proportion does the lift of the valve change with the speed? In Table 4 these results are tabulated and reduced to percentages, where they can be studied. The comparatively rough and uneven covering plate, holding the leather in place, is largely responsible for these erratic deviations. With

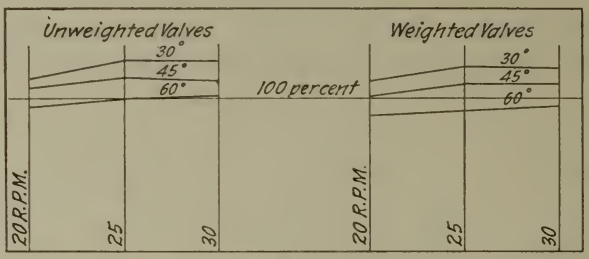

Fig. 16 Ratio of Agreement between Calculated and Actual Velocities through the Valve. See Col. 22, Table 3

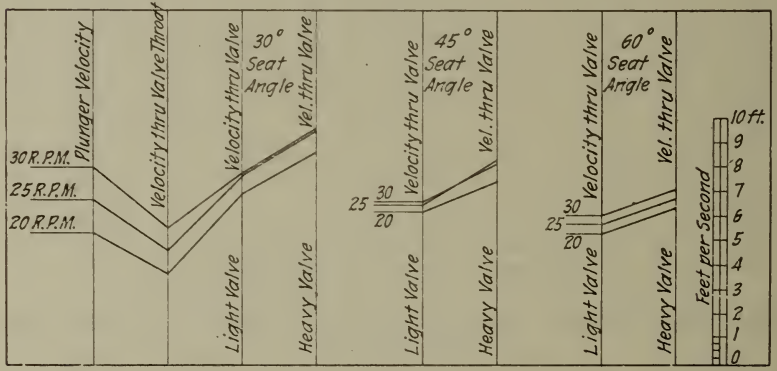

Fig. 17 Maximum Velocity, Feet per Second, of Plunger and of Flow through Throat of Valve and through Discharge Valve at 20, 25, and 30 R.P.M. and with 30, 45 and 60 Deg. Angle of Valve Seat

a smooth valve face, as shown in Fig. 10, the deviations would probably have been gradual and proportionate to the change of angle by the lift, and a value might be given to them for different seat angles, but that will not be attempted here.

42 Third: The speed remaining the same, how does a change of weight affect the lift of the valve? Table 5 in its first six columns groups together data previously given, and in column 7 is applied the theoretical law that the lift should be inversely pro-

portional to the square root of the weight of the valve. That is, taking the lighter valve and its lift, what should be the reduced lift by loading it? In every case, it will be observed, the valves lifted *higher* than they should, according to this rule. On the 30-deg. seat, this variation was only about 3 per cent; on the 45-deg. seat, about 5 per cent: and on the 60-deg. seat, about 7.6 per

TABLE 4 COMPARISON OF LIFT OF VALVE WITH INCREASED SPEED

THEORETICALLY, THE LIFT SHOULD BE DIRECTLY PROPORTIONAL TO THE SPEED

Seat Angle, Deg.	R.P.M.	Light or Unweighted Valve				Heavy or Weighted Valve			
		Actual, In.	Theoretical, In.	Deviation		Actual, In.	Theoretical, In.	Deviation	
				Per Cent	Per Cent			Per Cent	Per Cent
1	2	3	4	5	6	7	8	9	10
30	20	1.43	1.43	100	...	1.14	1.14	100	...
30	25	1.61	1.79	90	100	1.29	1.42	91	100
30	30	1.90	2.14	88	98	1.54	1.71	90	100
45	20	1.46	1.46	100	...	1.20	1.20	100	...
45	25	1.68	1.82	92	100	1.38	1.50	92	100
45	30	1.95	2.19	89	96	1.61	1.80	89	97
60	20	1.51	1.51	100	...	1.26	1.26	100	...
60	25	1.75	1.89	92	100	1.47	1.57	94	100
60	30	2.00	2.26	88	95	1.68	1.89	88	95
	40	2.50				2.50	47 revolutions		

NOTE.—These last valves would have gone to their stops at 40 and 47 r.p.m. respectively.

cent. Why should it have lifted higher than calculated? The only explanation seems to be that the increased weight reduced the angle of impingement of the issuing stream against the face of the valve, and that this change of angle was proportionately more on the 60-deg. seat than on the others.

NOTES

43 In Fig. 18 are shown the areas of the plunger displacement and the valve areas. At once one may ask why these enlarged valve areas are greater than that of the valve opening, or outlet; or perhaps more pertinently, why does not the engine

builder open the valve so as to give this required area when specifications call for, say, valve area of 145 per cent of plunger area? Simply because he cannot do it. If the valve was made light enough to open to this height, it would open, but it would close with a slam that would sound like a 10-in. gun explosion.

44 In Fig. 17 are plotted the plunger velocities and velocities through throat and outlet of valve, and it will be observed that

TABLE 5 COMPARISON OF WEIGHT AND LIFT OF VALVE

THE LIFT SHOULD BE INVERSELY PROPORTIONAL TO THE SQUARE ROOT OF ITS WEIGHT. DISCHARGE VALVE ONLY

Seat Angle, Deg.	R.P.M.	Weight per Sq. In.		Lift of Valve			
		Unweighted, Lb.	Weighted, Lb.	Unweighted, In.	Weighted, In.	According to Rule	Deviation, Per Cent
1	2	3	4	5	6	7	8
30	20	0.247	0.409	1.43	1.14	1.11	1.026
30	25	0.252	0.415	1.61	1.29	1.25	1.028
30	30	0.260	0.425	1.90	1.54	1.49	1.036
45	20	0.222	0.363	1.46	1.20	1.14	1.050
45	25	0.224	0.367	1.68	1.38	1.31	1.051
45	30	0.229	0.373	1.95	1.61	1.52	1.054
60	20	0.219	0.363	1.51	1.26	1.17	1.074
60	25	0.221	0.364	1.75	1.47	1.36	1.078
60	30	0.223	0.366	2.00	1.68	1.56	1.076
60	30	0.165	.....	2.50	....	....	.....

NOTE.—This last valve, weighing 0.165 lb. per sq. in. of seat area, making 30 r.p.m., would have gone to its 2½ in. stop.

with the lightest valve, on a 60-deg. seat, the velocity through the valve reached 5.30 ft. a second, while the plunger made 5.33 ft. a second. And yet, late waterworks specifications contain the clause that " the velocity through the valve shall not exceed 3 ft. a second "! Such low velocity was never obtained through a pump valve. It would call for a valve weighing only 0.06 lb. per sq. in. of seat area—1/10 of the actual pressure or weights of valves, in practice. The writer has weighed numerous valve springs on modern city pumping engines and found them to be about 0.60 lb. (more and less) per sq. in. of net seat area at the

start, and tightening up to as much as 1.50 lb. when lifted to a stop only 9/16 in. high, entailing a velocity of at least 9 ft. to 15 ft. a second.

45 Turning to valve card No. 41, it will be seen that it closes promptly even at 30 r.p.m. with a 2 in. lift, and its weight at the start was only 0.212 lb. per sq. in. and 0.227 lb. at its full 2½ in. lift, an increase of weight of only 7.1 per cent. This is

TABLE 6 VALVE RESISTANCE BY INDICATOR CARDS AND BY CALCULATION. SEE FIG. 19

Seat Angle, Deg.	R.P.M.	Unweighted		Weighted	
		By Indicator Per Cent	By Weight of Valve Per Cent	By Indicator Per Cent	By Weight of Valve Per Cent
1	2	3	4	5	6
30	20	2.75	2.31	4.10	3.83
30	25	3.09	2.37	3.76	3.89
30	30	4.29	2.38	2.91	3.98
45	20	2.85	2.08	3.79	3.40
45	25	3.46	2.10	4.12	3.44
45	30	4.17	2.15	4.49	3.50
60	20	2.95	2.05	3.45	3.40
60	25	3.27	2.07	2.95	3.41
60	30	2.63	2.09	3.44	3.43
Averages.....		3.27	2.18	3.67	3.59

an initial weight of only about one-third and a final weight of only about one-sixth as much as is put into spring-operated pump valves.

VALVE RESISTANCE

46 It will be noted that the pressure line in both suction and discharge chambers was traced on the pump indicator cards. From these, the valve resistance commonly called friction could have been obtained if an average of many cards had been obtainable; but those taken for each experiment were too few to give rational results. In Table 6, columns 3 and 5 are copied from Mr. Mayer's records and plotted in Fig. 19. The fluctuations are too great for any reasonable deductions.

47 The writer has therefore attempted to make a calculation of this resistance based upon the weight of the valve which, it seems to him, is the source of this loss. Practically, all other resistances within a pump are of relatively small amount, and

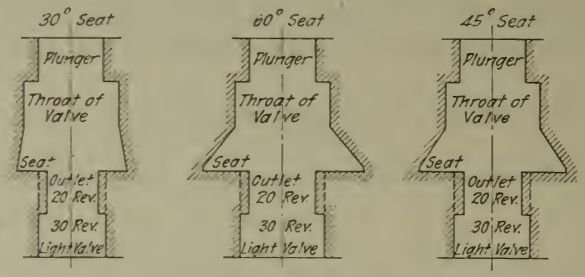

Fig. 18 Proportional Areas at Maximum Velocities

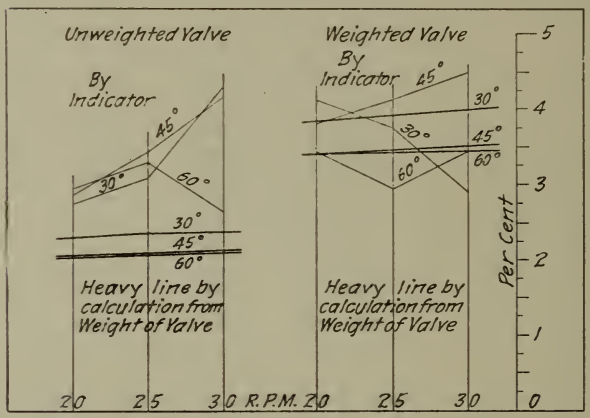

Fig. 19 Percentage of Valve Friction to 72-Ft. Head

hence the weight of a valve furnishes a reasonable criterion of pump losses.

48 It will be recalled that the apparent velocity through the valves is given in column 17, Table 3, and that no further allusion was made to the effect of contraction through the valve. If a calculation for loss of power is to be made, its amount will depend upon the amount of contraction, for the velocity must then be increased in an inverse ratio to the contraction. For the purpose of illustrating the method followed, a contraction of 81

per cent has been assumed. That would increase the velocity 23 per cent and the velocity head 50 per cent. Taking the weight of the valve as given in column 20 and increasing it by 50 per cent, gives a fair basis for calculating the loss of power within the pump. To bring this to a percentage basis it must be multiplied by two to apply it to both strokes and divided by 32 lb. for short (31.74 lb.), being the net total head in column 5, Table 1, and these values are given in columns 4 and 6, Table 6, and plotted in Fig. 19.

49 It must be left to the reader to decide whether the writer's assumptions and deductions are rational and justifiable.

50 The writer wishes to express his obligations to Mr. E. G. Grace, General Manager, Bethlehem Steel Company, for permission to make public the technical information obtained by these experiments.

No. 1367

DEVELOPMENTS IN MACHINE SHOP PRACTICE DURING THE LAST DECADE

REPORT OF SUB-COMMITTEE ON MACHINE SHOP PRACTICE

In this report the principal improvements in machine shop practice developed approximately during the last ten years are reviewed. The reason for presenting an extended review now, is that the sub-committee on machine shop practice should record the advances made each year, but in the present case the committee having been only recently organized, there exists in the Society's proceedings no consecutive account of progress. To carry the record back to a logical date, that of the report made by Fred J. Miller to the Bureau of the Census, in 1905, is mentioned but, of course, the actual date is somewhat indefinite.

2 The review is incomplete because of the limitation of time to prepare it, and because of a natural diversity of opinion as to the limitations of what should properly be considered machine shop practice. If this review has included matters generally regarded as outside of the limitations, it may be considered that the larger field was included because of practical difficulties in drawing a well-defined line between machine shop practice as such and machine design, administration, heat treatment of steel and other factors affecting production. To ignore them is impossible. The operation of metal-working tools and their productive capacity are as much matters of machine design, arrangement and management as of individual skill. In fact, productive capacity in the past depended very largely on the skill and initiative of the workman, but now the rate of production in many branches of manufacture is almost entirely governed by the construction of machines, so that the designer, when viewed in this light, is closely allied with shop practice.

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"The transference of skill" by the machine designer from the operators to the machines has embodied in the latter much of the accumulated experience of many mechanics working on simpler and more primitive tools. So it may be asserted that much of the improvement in machine shop practice has appeared in the improvement of machine tools themselves in one way or another.

3 *Variety of Machine Shop Practice.* When it is considered that machine shop practice embraces a very wide variety of work, ranging from the smallest to the largest manufactured products, and all kinds of repair work, the difficulty of classifying and differentiating it can be appreciated. Locomotive repair work, for example, is in a class by itself, one in which peculiar methods have been developed, differing radically from those followed in doing similar work in other lines. The slotter, for example, is a machine tool in high favor in railroad shops and, outside of shipyards, has been little used elsewhere. It is, however, slowly coming into use in manufacturing works. The milling machine until a few years ago had not gained a foothold in railroad repair shops, largely because it was, and still is, not so well adapted to handling the general run of plane finishing required in these shops as the planer and shaper. The latter are adaptable machines peculiarly suited to non-repetitive operations, while the milling machine excels in manufacturing.

4 *Increased Weight of Machines.* One of the noticeable changes in machine shop equipment during the past ten years, is the increased weight and power of machine tools, which was made necessary through the general application of high-speed steel cutting tools. While the increased capacity of machines which followed in the wake of the application of high-speed steel was directly instrumental in bringing about such increases in the weight of machines, the result has been beneficial, not only in increased production but in accuracy of product, because of greater rigidity and consequent better preservation of alignment.

5 *Foundations.* Progress has also been made in the methods of erecting machinery. With large size machines the foundation plays a most important part in the accuracy of the work produced. For instance, the provision of a massive, well-built foundation for large planers is imperative. When not provided,

the weight of the machine and work must inevitably cause settling and springing out of shape. Every mechanic knows that such a disturbance of the foundation affects the alignment, which means that the work produced on the planer will be inaccurate; the use of leveling plates under the feet of planers to provide uniform support and means for correcting errors of alignment has become general; the elimination of defective action due to want of support, and the increased mass and rigidity have in the case of planers resulted in the production of plane surfaces closely approaching theoretical planes when the work itself affords the necessary rigidity and uniformity of section. Hence, hand-scraping to perfect alignment is not as necessary as it was a few years ago, and it may be said that on some classes of work scraping is now largely more of an ornamental than a corrective process. On high-grade machines, however, scraping or grinding probably will continue to be necessary for correcting inaccuracies of surface and alignment.

6 *Electric Motor Drive.* The subject of electric motor drives was given much prominence in discussions of machine shop practice a few years ago, the bias being generally in favor of individual motor drive. While subsequent experience has shown that the direct-connected electric motor has certain limitations and disadvantages for some classes of work, it has also shown that the same results and efficiency cannot be obtained with the group system of driving modern designs of machine tools. The investment in motors is less with the group system, but the flexibility of the independent drive is lost. When a machine driven by an independent motor is properly equipped for easy handling, increase of production over that of the same machine in a group system is practically insured. One machine tool builder has found it a good rule to apply independent motors wherever the power requirement is 5 h.p. or more. Hardened steel gears in the driving train reduce gear troubles to a minimum. When the character of work is constantly changing and new groupings of machines are required from time to time, the individual drive is at great advantage. The machines can then be placed "like men on a checkerboard."

7 Improvements have been made in the characteristics of variable speed and reversible motors. Automatic controllers

have been introduced which make the stopping and starting of machines simply the matter of pushing a button. These devices protect the motors, save time and protect the machines from shocks.

8 *Automatic Machines.* Progress has been made in the development of automatic and semi-automatic machines for the production of duplicate parts from bar stock, iron and brass castings. Many special fixtures have been designed for automatic screw machines which have widened their scope to include a great variety of small parts on which a number of operations must be performed before cutting off from the bar, such as small spiral gears, capstan screws, etc. The cost of production is so greatly reduced that these machines are generally applied where the quantity of work to be produced warrants the cost of installation and tool maintenance. The scope of other types of automatic machine tools is wide and constantly increasing. Certain examples will be mentioned later. Many pages could be devoted to the improvements in these machines alone without exhausting the topic; but attention is called here to the simplification of equipment required for the manufacture of highly developed specialties because of the possibility of buying the bulk of the parts from screw machine product specialists. A manufacturer of a specialty can, if he chooses, organize on an assembling basis, confining his actual manufacture to the making of a few principal parts. This development is having a marked influence on our export trade.

9 *Training Mechanics.* Modern methods of manufacturing are responsible for limiting the employment of men to specific operations only, and pursuit of the plan is making it hard to secure all-round machinists. Young men come into a factory and soon acquire the skill necessary to become proficient drill press operators or milling machine operators. They are able to earn fairly good wages in a shorter space of time than if they served the necessary term of apprenticeship to become competent all-round machinists. The endeavor has been made in the larger manufacturing cities to offset the narrowing effect of specialized training by the introduction of training schools in which young mechanics are given instruction in the theoretical side of their work. The results obtained by this training are beneficial. Men are acquiring both theoretical and practical knowledge, so that to a limited extent at least, they are obtain-

ing the benefits of a technical education coupled with practical training.

10 *Scientific Management.* One of the noteworthy improvements in modern shop practice is the application of *the principles* of scientific management. While the details and elements that make up the complete system advocated by Mr. Taylor are not widely used in full, undoubtedly the fundamental principles are coming to be more thoroughly understood and are being quite generally applied. Some of the more specific improvements referred to in the following may not be strictly modern, but doubtless most of them can be so classified because they are now common, whereas ten years ago they were found in comparatively few shops.

11 *Standardization of Tool Grinding.* The turning and planing tools in many shops are now ground to standard shapes in the tool room on special tool grinding machines. This method not only saves the time of the machine tool operator, but insures the grinding of all tools to the correct form and it also saves what is of greater value, the loss of the productive time of the machine tool. The stock of tools required for a shop is also much smaller.

12 *Checking System for Small Tools.* Most shops at the present time keep all small tools in the store room, except those in actual use. In this way, the tools are kept in good condition and a much smaller stock is required. Not many years ago it was quite common for each workman to have his own stock of tools. Consequently there were good tools and poor tools, just in proportion as there are good workmen and bad workmen. For instance, a lathe operator had his own stock of mandrels which were made to suit each particular job, and when necessary one man borrowed from another. The result was that the workmen wasted a great deal of time wandering around looking for mandrels and other auxiliary equipment. It may seem somewhat out of place to refer to this well-known method of handling small tools by a checking system in a paper on modern improvements, because it is an old feature of many shops. This system was not generally employed ten years ago in many of the small and medium sized shops.

13 In some shops in which scientific management has been applied, and where the work in connection with each operation is planned in advance, it is the practice to make up, at the time the

work is planned, a list of the tools that will be required for the performance of each operation, this list being sent to the tool room in advance of the time that the tools will be required, and the tools delivered to the workman by a tool carrier, thus saving the time of the workman as well as that of his machine.

14 *Speeds and Feeds.* The important questions of speed and feed in connection with machining operations have, during recent years, been carefully studied by many superintendents and foremen who formerly relied entirely on the judgment of the workmen, which was sometimes good and sometimes bad. Speeds and feeds are fixed in the planning department and are based on the power of the machine and character of the metal.

15 *Tool Selection.* More thought is now given to the selection of machine tools for various classes of work. A few years ago, many shop foremen used a certain type of machine for a given class of work, because that type had always been used for that purpose. It seems that there is much less conservatism today and precedent does not count for as much as it did a few years ago. The tendency is to design the machine for the work instead of adapting the work to the machine equipment.

16 *Inspection of Finished Parts.* Machine parts are now carefully inspected in many shops by an independent inspection department, whereas formerly the only check on the accuracy of machined work was in the erecting department; if everything could be assembled, the work was considered satisfactory. The more exacting requirements of recent years, however, made necessary the general adoption of a rigorous system of inspection. Thorough inspection and testing are necessary features of interchangeable manufacturing methods; both tools and system have become highly specialized. In the manufacture of firearms and other high-grade products, from fifty to one hundred separate tests or gagings on a single part are not uncommon.

17 In some shops doing work of a miscellaneous character, it is the practice to have an inspector go to the machine at the time a job is started to instruct the workman as to the degree of accuracy, the kind of finish, etc., and to make sure that he understands all the requirements with reference to the quality of the work and that he does not make a mistake in reading his drawing, setting his measuring tools, etc. The inspector inspects the work done in the operation in question upon the first piece where there are several pieces to be done, thus avoiding the danger of

an error in the first piece being repeated on all subsequent pieces. An inspection is also made upon the completion of each operation on all pieces in the lot for the purpose of detecting any error that may have been made, so that steps may be taken for its correction without delay.

18 *Cooling Lubricants.* Another improvement in shop practice worthy of mention is the more generous use of cooling mediums on cutting tools, and the provision of distribution systems for both the tools and the bearings supplied from a central reservoir.

19 *Use of Multiple Tools.* The increasing use of tools in multiple is a feature that is noticeable in connection with the modern machine shop. Milling machines are now quite extensively used having several spindles, making possible the finishing of three or even five sides of a casting at one setting; many holes are drilled simultaneously and in practically the same time formerly required for drilling one hole, and so on.

20 *Better Facilities for Setting up Work.* The up-to-date foreman or superintendent has come to realize that the efficiency of his shop depends largely on the amount of attention given to the little things. The furnishing of proper clamps and bolts to machine tool operators is a case in point. Poor bolts and warped clamps around a machine often double the time required for setting up work. At the present time, most shops are equipped with proper clamping facilities and, in many cases, clamps of the right shape and bolts of the right length are furnished for each particular job, along with the other tools, such as cutters, reamers, etc.

21 These tools are being kept in the toolroom instead of being kept at the machines as formerly; thus the responsibility for their maintenance in first-class condition is concentrated in the tool room, where they are inspected upon being returned after their use in connection with each job, and any necessary repairs promptly made. The difficulty of holding a large number of men responsible for the upkeep of these tools is thus avoided, and each workman has a larger variety of bolts, clamps, blocks, etc., to draw upon than he can possibly have where each man has his own supply.

22 *Location of Tools and Lighting.* More attention is given to the relative location of tools in the shop, in order that all machining operations can be performed with no unnecessary

handling of the work, the aim being to finish parts by advancing them from one tool to another in a direct line without any see-sawing or useless movements. Shops are constructed so that the light will be properly diffused, which makes it easier for the machinist to do accurate work and reduces the amount of spoiled work. The physical comfort of the workmen is also receiving more and more attention.

23 *Drilling Machines.* High-speed steel has been substituted for carbon steel in the manufacture of twist and flat twist drills with marked improvement in capacity. The speed of drilling has been increased two or three times, if not more. The drilling machine, the most common machine tool, has been redesigned to meet the new requirements and so rapid are some of the most highly developed machines on the market, that holes can be drilled in boiler plate more quickly than they can be punched.

24 The claim is made by engineers who have carefully investigated the matter, that 80 per cent of all the holes drilled in general manufacturing are three-quarters inch in diameter or less. The fact is pointed out that general purpose drills with a capacity of from one-eighth inch to three inches are not efficient except within a narrow zone somewhere between the two extremes cited, desirable as they are when on deep drilling or on large diameters requiring a geared drive; hence the general recognition of the importance of comparatively small capacity drilling machines in the equipment of efficient plants.

25 Multiple drilling machines with from two to fifty or more drills in operation simultaneously, have been developed. Inverted drilling machines with which the capacity of drills is increased largely because of the rapid clearing of chips from the holes are being used with satisfaction, also multiple drilling machines working in several planes simultaneously, making feasible the drilling and reaming of all holes in machine parts such as automobile cylinders at one operation. Another development is the rotary table semi-automatic drilling machines which relieve the operator of lifting and lowering the spindle and engaging and disengaging the feed.

26 A minor improvement of machine shop equipment which has increased the productive capacity of single-spindle drilling machines is the so-called roller grip drill chuck which enables the operator to insert and remove drills, reamers, counterbores and other tools without stopping the spindle. These chucks

are useful for certain kinds of work requiring drilling, reaming and counterboring operations. In the same connection, mention should be made of tapping attachments, stud setting chucks, drill speeders, and similar attachments which improve the working range of drilling machines.

27 The need of the old style reversing countershaft has been eliminated from drilling machines by automatically reversing tapping attachments, automatically opening stud-setters; and many screw-threading operations are now performed on drill-presses with special forms of automatically opening dies, at a great saving over the former method of chucking the work on a turret lathe.

28 *Lathes.* The improvements in lathes have been chiefly in matters of strength, power, and details of construction. Spindles are made larger and are supported in longer bearings, improvements have been made in carriages, tailstocks, tool-rests, apron mechanism, stops, ways, spindle noses, gear boxes, change gear and feed mechanisms, etc. Three-stop cone pulley and double back gear headstocks have been widely adopted for high-power rapid reduction lathes, this construction providing a wide range of speeds with simple and efficient mechanism. Quick change gear mechanisms which enable the selection of gear combinations for screw-thread cutting to be made practically instantaneously have been generally provided by lathe builders. The demand for simple lathes, that is lathes without change gears, lead screws and other standard features of complete engine lathes, has not been insistent. Hence we still have the anomaly of thousands of lathes in manufacturing plants that are never used for thread cutting but which have the change gears, lead-screws and other parts required for screw cutting.

29 The reason ascribed by one prominent lathe builder is that the application of the simpler mechanism required for feeding only, makes comparatively little difference in the cost of a lathe. The elimination of the change gears reduces the number of available feeds. But the chief objection is the loss of inventory value when regarded as second-hand tools.

30 *Wheel Lathes.* A remarkable increase of efficiency has been made in locomotive driving wheel and carwheel lathes. Ten years ago, turning the tires of two pairs of drivers was a good day's work, and most railway shops did much less. High-speed steel has worked perhaps one of its greatest triumphs in

this field. Tire-turning lathes have been developed to equal the capacity of the best steels and the productive capacity has been increased to eight or ten pairs of drivers a day. The increase of capacity of steel tire carwheel lathes is nearly as great, having been raised from five or six pairs to eighteen or twenty pairs a day.

31 *Planer Drives.* A weak feature of planers and a serious limitation to the power of large planers, is the common shifting belt reversing mechanism. On small planers even the shifting belts are objectionable because of slipping and the characteristic squeaking noises, but on large planers their fault is of more serious nature. Wide belts necessary to transmit the required power at practical speeds cannot be quickly shifted from the tight to the loose pulleys and *vice versa.* Clutches of various designs are used with varying degrees of success. The reversing electric motor direct connected to the planer drive has been developed to the point that makes its success assured. Probably this change of drive is the most marked improvement in planers made in recent years. Many details have been improved and the greater strength and rigidity of the modern planer coupled with excellent workmanship insures accurate work.

32 *Milling Machines.* A change in the machines for producing plane surfaces has been going on gradually but surely. Planers, shapers and slotting machines are being displaced by various types of milling machines in manufacturing plants. As plants change from a building to a manufacturing basis, the superiority of the milling machine as a manufacturing machine gives it the preference. The development of coarse-pitch teeth milling cutters and of face milling machines, are two of the marked improvements.

33 Single pulley drive with which the maximum power capacity of a given width belt can be transmitted to the machine irrespective of the work spindle speed, and geared speed boxes giving a wide range of positive speeds are other important changes in design. The vertical spindle milling machine has been developed to a high plane of efficiency, especially for small work. Rotary table machines with quick action clamping devices revolving continually while the operator places the work in position and removes the finished parts, are coming into extensive use.

34 Characteristics of modern milling machines also common

to other modern machine tools are convenience and ease of operation, starting and stopping levers being placed where the operator can grasp them without leaving his normal position; speed and feed levers so placed as to permit the operator to control the rate of cutting while watching the work; power traverse of slides on heavy machines; etc.

35 *Boring Mills and Boring Machines.* The boring mill has taken a commanding place as a machine tool for both light and heavy work. When equipped with turret heads and a proper complement of tools its productive capacity has been made second to none. Convenience of operation, economy of floor space, compactness of design, adaptability to use of lubricants on cutting tools, are some of the advantages of this machine which have been emphasized in the new designs brought out in recent years. The builders have studied to find the best order of operations and equip with chucks and tools that produce an output far in advance of that possible ten years ago.

36 The horizontal boring machine of the bed and carriage type which has been developed practically within the past ten years, partakes of the characteristics of the lathe, milling machine and boring mill, and in the improved designs is superior to any one, for certain classes of work. One of many uses to which it is devoted is that of boring jigs and fixtures for interchangeable production. Not only is it useful for jig making but it is peculiarly well suited for manufacturing machine parts interchangeably without the use of jigs. This important fact enables machine tools and similar high-grade machines to be economically produced on the interchangeable plan during the very active period of developing the design.

37 *Magnetic Chucks.* Magnetism for holding steel and iron parts for grinding, planing and turning operations has been made useful, especially for thin parts that are easily sprung out of shape by ordinary clamping means. Magnetic grinding, planing and lathe chucks have come into common use in plants having up-to-date equipment.

38 *Grinding Machines.* The surface grinder, especially the vertical spindle type, has made great strides during the last few years. The improvements in cylindrical grinding methods are also worthy of mention, especially the use of the heavy-duty grinder for removing stock formerly removed by a second or finishing cut in a lathe. Grinding stock from the rough as

in the finishing of drop forged crankshafts is also a good example of modern practice. The stage has been reached in cylindrical grinding practice which places the cylindrical grinding machine on a coördinate basis with the lathe as a tool for finishing cylindrical work. A feature of cylindrical grinding machines which has greatly increased productive capacity in multiple diameter work is the stop bar or semi-automatic measuring attachment which enables shouldered shafts, etc., to be duplicated without measurement. Machines for internal grinding of non-revolving work with planetary spindles provides for the economical and accurate sizing of engine cylinders and other parts difficult to rotate. Vertical grinding machines and rotary work-tables utilizing magnetic chucks or other quick action clamping means for quickly securing and releasing the parts to be ground have come into extensive use for high-grade interchangeable manufacturing. The development of the disk grinder from a mere smoothing machine to a powerful machine tool of great capacity for finishing plane and curved castings from the rough is one of the most interesting phases of modern practice.

39 *Screw-Thread Milling.* One of the notable improvements in the cutting of lead screws, feed screws and other machine screws, especially those required to be exact in pitch and lead is the substitution of special milling machines in place of the common engine or screw cutting lathe. Milling screws is by no means a new idea but the commercial development of special screw thread milling machines has taken place within a few years past. The machines not only increase accuracy but materially reduce the labor cost, one man being able to tend from three to six machines.

40 *Jigs and Fixtures.* The development of jigs and fixtures for interchangeable manufacturing has been remarkable. The expansion of automobile manufacture has been enormous and most of the leading concerns employ jigs and fixtures exclusively, thus insuring interchangeability, low production cost and systematic production. Many improvements have been made in the way of clamping devices, standardization of bushings, handles, levers, frames, etc., too numerous to mention specifically. Toolmaking has been developed on manufacturing lines, and in fact, several concerns specialize on the making of tools, jigs, fixtures, punches and dies, and produce them exclusively for manufacturing plants.

41 *Gear Making.* The demands of automobile users for quiet running gears have imposed on gear makers conditions very difficult to meet. The limits of error in shape and spacing of teeth have been greatly reduced, and coupled with the necessity of making highly accurate gears, often has been that of producing them from tough alloy steels, heat treated after cutting the teeth. Hardened and case-hardened gears are also demanded with minimum limits of error. The planing process has been generally substituted for milling in the manufacture of high-grade bevel gears, thus producing theoretically correct tooth shapes instead of the approximation possible only by the formed milling cutter process.

42 Gear hobbing machines have been widely adopted for cutting spur and spiral gears, because of the greatly increased capacity and simplicity of operation. Machines and methods of cutting cheaply the teeth of integral herringbone gears, have been developed, thus making this form of gear generally available for machinery in which it has not been commonly used because of high cost of the two-part type. The demand of automobile users for noiseless gears has led to grinding the teeth of gears after hardening to correct form, and form grinding generally, including splined shafts for automobile change gear shafts and similar parts. Progress has been effected in the making of gears by the hot rolling process, producing a product of exceptional physical characteristics with a minimum waste of material. Progress has been made in the use of rotary cutters in gear manufacture. The substitution of high-speed steel for carbon steel has, of course, made higher cutting speeds possible, and improved designs of gear cutting machines have in turn made the cutters work more efficiently because of greater power and rigidity of support. Exhausts to draw away the chips and cool the cutters have made a notable improvement in efficiency, making greater speed and smoother cutting possible.

43 *Small Tools.* The design and manufacture of small tools have been improved principally in details. There has been a tendency towards decreasing the number of teeth of milling cutters, making possible heavier roughing cuts with proportionately less expenditure of power. While the extremely coarse spacing of the teeth advocated by some leading engineers has not been generally adopted, all manufacturers of milling cutters have, to some extent, modified the tooth spacing of their cutters. Several

special milling cutters with inserted teeth have also been proposed and, to some extent, adopted. The tendency has been towards the design of easily adjusted removable blade type reamers, and several satisfactory designs have been brought out. The flat twisted high-speed steel drill has been commercially developed, and several modifications of it have been placed on the market. The main progress in tap making, perhaps, has been towards a more satisfactory shape of the flute, and several tap makers have made experiments in this direction.

44 In general, there has been an attempt to design inserted cutter tools to a greater extent than formerly. This activity has been prompted largely by the high price of high-speed steel which has made it necessary to make the cutters only from this material, while the body and shanks of the tools are made of machine steel. Many experiments have been made to find the steels best adapted for a given service and the best heat treatment for the selected steels. Improvements have been made in the tangs of twist and flat drills adding strength sufficient to enable them to stand up to the requirements of high-speed drilling and drilling machines.

45 *Micrometers and Gages.* During the past ten years, the micrometer has become a measuring tool of general use. Almost every mechanic employed in up-to-date shops, who owns a kit of tools, regards a "mike" to be as necessary as calipers. Many shops also supply them from the tool rooms on the workman's checks. As the result of the general use of the micrometer, one-thousandth inch has become the unit of measurement for fixing limits, and a limit of one-quarter thousandth is not uncommon in grinding operations. Not only is there a great advantage in the matter of accuracy in the use of micrometers over the old style of calipers set to a scale, but there is a considerable saving in time through their use, both in the matter of setting to size and in eliminating the element of uncertainty, permitting the workman to proceed with his work with greater confidence. So general has the use of this unit become, and so convenient is it for small dimensions, that a strong movement is under way to make it the standard for measuring wire, sheet metals, etc., in place of the irrational and arbitrary wire and sheet metal gages. One of the largest electrical concerns abandoned the wire and sheet metal gage systems in 1904 and adopted the decimal system in which one thousandth inch is the limit for use in all specifications of wire, sheet metal, fiber, etc.

46 Standard length gages have been improved. The so-called Swedish gages are furnished under guarantee of an error of not more than one one-hundred thousandth inch, per inch length, this limit being proportional to the length. Thus a gage one-tenth inch long has an error limit of one-millionth inch, and so on. Hence a stack of these gages presents an over-all length accurate to limit of one one-hundred-thousandth inch per inch of length.

47 *Screw Threads.* The so-called United States or Sellers standard screw threads have come into general use for machine construction during the past twenty or twenty-five years and have proved generally satisfactory for locomotives, cars, steam engines, and other typical heavy machinery but the automobile brought a new condition for which the standard threads were not well fitted. It was found that the jar and vibration to which motor cars are subjected, speedily loosened the comparatively coarse thread screws and nuts threaded to these pitches, and the need of finer pitches was soon recognized. We now have the finer screw pitches established by the Society of Automobile Engineers, and the merits of these pitches have won them recognition in other lines of machinery where for constructive reasons, pitches finer than the United States standard were desirable.

48 The A. S. M. E. standards for machine screws, which were adopted after a thorough investigation of the problem, should be mentioned also, in this connection. No work that this Society has done is of greater importance than that of fixing on standards of screw threads to be recommended to the engineering profession.

49 *Portable Tools.* An important factor in shop work is the portable electric motor drilling and reaming machine which has been developed in several efficient forms during the last decade. These portable machines, supplemented with the pneumatic riveting, chipping and chalking hammers, have virtually revolutionized shipbuilding, bridge-building, and structural steel work, and in the erection shop have wrought great changes, making possible rapid and efficient machine work on the parts as erected and thus saving labor and time through the elimination of transport to and from stationary machines.

50 *Power Hack Saws.* The first power hack saw machine was put on the market about twenty-five years ago. During the past few years the range and capacity of these machines has been greatly extended. They are furnished with quick return

stroke, relief action for back stroke, means for using the full length of the blade, angle chucks and other improvements that have made this humble machine a real machine tool of high efficiency and low operating cost. Blades of great durability are furnished at low cost.

51 *Brass Working.* The changes in brass working are principally in the development of automatic and semi-automatic machines for machining standard parts made in large quantities. Some of the machines have effected great reductions in cost, in some cases reducing the labor cost to one-fifth, or less, of that before the change was made. Carbon steel tools still have the call in some plants in preference to high-speed steel tools; carbon steel holds a keener edge and is generally more satisfactory for working the brass used in the manufacture of globe valves, cocks, etc. It is claimed, however, that much of the difficulty experienced in using high-speed steel on work of this character is due to improper heat treatment of the high-speed steel, and this contention seems to be borne out in practice as certain shops doing brass work of this character are using high-speed steel exclusively.

52 Chucks operated by compressed air have made a great improvement in the operating efficiency of brass lathes. The work can be chucked and removed without stopping the work spindle, thus saving the time and strength of the operator. The development of the air chuck has taken place chiefly within the past few years, and it is one of the principal improvements made in hand operated turret lathes.

53 *Manufacture of Guns and Pistols.* The methods of making guns, pistols and similar high-grade products have been revolutionized. A representative of one of the largest machine tool building concerns estimates that the efficiency of workmen in these lines has been doubled by the introduction of improved machinery and methods. When the fact is considered that high-grade gun parts are made interchangeable, the limits of error allowed being very small indeed, and that gunmaking is one of the oldest industries, the importance of the improvements made can be better appreciated.

54 *Pressworking.* Pressworking machines, punches, dies and other tools, have developed to an extent not generally realized by engineers in lines not affected. Here again the development of highly specialized methods of manufacturing automobiles has had a marked influence. Crank presses of one thousand tons capacity are in use. Pressed metal forms of large size are

a commonplace. But not only has the development been remarkable in point of size, but also in detail of manipulation. Parts requiring several operations, for example, are turned out by multiple plunger presses, one finished piece being formed at every stroke. An interesting feature of presswork is that shapes are produced impossible of duplication by any other known process. In this respect, presswork differs from most other machining processes; it is possible to duplicate, for example, lathe or planer work by hand tools, but not so many drawing and forming operations in daily use. A branch of presswork little known is the extrusion of shells, tubes, collapsible tubes, etc., of copper, brass and alloys. Extrusion of bars in many shapes used in manufacturing locks, small machines, etc., has become a large industry which has been largely developed in recent years. The possibilities of extrusion are practically unlimited, and this process in common with other metal working methods extensively used has an important effect on the practice of shops using the shapes in their products.

55 *Heat Treatment.* The advent of the high-speed steels and the alloy steels used in automobile gears, shafts and other high-grade machinery subjected to shocks, has necessitated the development of furnaces and equipment for the scientific heat treatment of steel. Accurate determinations of quenching and annealing temperatures are absolutely necessary, and the pyrometer has become an indispensable instrument in the furnace room. Gas and oil furnaces have been greatly improved in the matters of efficient use of fuel and control of temperatures.

56 *Die Casting.* The production of alloy castings in metal dies has grown to large proportions. Gears, bushings, bearings, type wheels, machine parts, covers, etc., are produced accurate to pattern and free from fins. This phase of manufacturing machine parts has a decided bearing on machine design. The extension of die casting to include not only the low melting alloys but also brass and cast iron, promises to make decided changes in some lines. Cast-iron gears true to pattern on which machine work is reduced to the boring of the hub for the shaft, may become common in agricultural and similar grades of machinery.

57 *Ball Burnishing.* Finishing processes are important in giving attractiveness to manufactured goods. Hand methods are slow and costly and the natural tendency is to use automatic mechanical methods for finishing as well as for making. One

process of considerable interest because of its simplicity and effectiveness is the steel ball burnishing method, consisting essentially of a tumbling barrel partly filled with the articles to be finished and a mass of hardened steel balls of various sizes.

58 *Thermit Welding.* An effective process has been introduced extensively during the period covered, for welding heavy sections of steel and iron. The thermit process is being successfully used for welding broken locomotive frames, cylinders, crankshafts, propeller shafts up to any thickness and diameter. The fractures are soundly united by a steel bond applied *in situ* at a fraction of the cost of removing the parts, welding them in the smith shop, finishing in the machine shop and replacing. Hundreds of locomotive frames have been welded by this process with almost uniform satisfaction and at a saving.

59 *Autogenous Flame Welding and Cutting.* The development of oxy-acetylene and oxy-hydrogen welding and cutting torches has given the industrial world truly remarkable tools for building up or cutting apart. The welding of sheet metals is accomplished with a speed and evenness hardly possible in any other process, and the cutting of steel or wrought iron is done close to the line and with such rapidity as to equal or exceed the performance of any machine tool adapted to the same work. Recent improvements make possible the cutting of all manner of shapes from steel plates up to six inches thickness so close to the outline of the pattern that a light finishing cut suffices to transform the piece into a finished die.

60 *Cranes, Hoists and Trolleys.* The general provision of traveling cranes, jib cranes, hoists and trolleys in modern shops has made a great change in the manner of handling work and the attitude of mechanics to big jobs. When a casting weighing many tons had to be machined in the old-time shop it was a herculean job to transport it from one machine to another and fix it in position. So general has the power traveling crane become in the past few years that it is a commonplace sight to see machine parts weighing many tons being transported through shops and placed on machines with no fuss and requiring the help of only one or two men besides the crane operator.

61 *Accident Prevention.* The prevention of industrial accidents has received wide attention in industrial establishments. In machine shops accident prevention is a relatively simple matter, but there are a few points of danger which have become generally appreciated, and all of the better types of machines are

equipped with guards to cover gearings, set screws, flywheels and all moving parts not necessarily exposed. There are also certain sources of danger which exist in practically all manufacturing plants and the general rules to prevent accidents from slipping ladders, piles of castings, elevators, etc., are being applied.

62 Time and space will not permit going much more into detail, and the following are enumerated without elaboration: Introduction of ball bearings in the construction of machine tools, effecting a saving of power, increasing the efficiency of operation, and capacity through reduction of bearing widths; development of cam grinding machinery and attachments making possible the rapid and economical production of camshafts having cams integral with the shaft; development of the dynamic balancing machine and the general recognition of the need of dynamic balancing for high-speed revolving parts; artificial production of highly efficient abrasives or cutting particles for grinding wheels, and improvements in grinding wheel manufacture; development of improved lubricating systems for machine tool bearings; development of machine tools to their utmost capacity by providing means for working the tools all the time, employing extra men to prepare work and also by tooling the machines to bring them up to their highest productive capacity; development of spline milling machines for cutting keyways, key slots, drift holes, cam grooves, recesses, etc.; finishing square, hexagon and other shaped holes by the broaching process, insuring interchangeability and rapid production at low cost; general improvement in sanitary conditions of shops, improved lighting facilities both by day and by night through the use of saw-tooth roofs, large windows fitted with metal window frames, improved electric lights and means of distribution; provision of lockers; and the establishment of shop restaurants in localities that formerly had nothing better than the corner saloon.

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No. 1368

THE VAUCLAIN DRILL

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Speaking generally, there can be no better definition of economical drilling than "rapid drilling"—the saving of time. The fact that a drill will cut at some phenomenal speed or will consume such and such an amount of power means nothing so far as productive capacity is concerned. The object in view is the removal of chips.

2 The productive possibilities of any metal-cutting tool are limited by its stress and heat resisting capacities. The temperature at which it will continue to operate successfully depends upon the excellence of material and manufacture, but the rapidity of heat generation and the stresses set up in the tool depend upon the design of the tool and the selection of feeds and speeds.

3 With a given excellence of material and manufacture, the strength of the tool must depend upon the extent and disposition of the section. The size of the drill is necessarily limited by the size of the hole to be drilled. The section of the drill has a lesser area than that of the hole, since space must be provided for the discharge of chips from the hole. The design of the drill should therefore be that which will give it the maximum of strength and strength conservation.

4 Cutting stress is practically independent of the cutting speed and with a given feed is proportional to the lip angle of

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the cutting edge. The cutting stress does not increase as rapidly as the feed.

5 The rapidity with which cutting heat is generated depends upon the cutting speed, the depth of feed and the lip angle of the cutting edge.

6 Since both the stress and heat are influenced by the keenness of the cutting-edge, it is desirable that the lip angle be as small as possible. But it must be blunt enough to carry off the

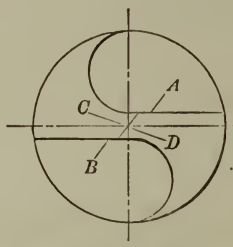


FIG. 1 COMMON TYPE OF TWIST DRILL

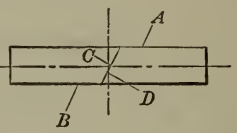


FIG. 2 COMMON TYPE OF FLAT DRILL

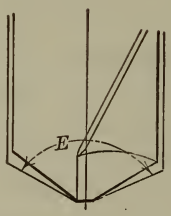


FIG. 3 SHOWING INCLUDED ANGLE OF CUTTING EDGES OF 118 DEG.

heat and to support the chip pressure, which falls more or less back of the actual cutting edge, according to the depth of feed. Since the chip is torn, not cut from the work, rupture between work and chip precedes the actual cutting edge. The heavier the feed, the farther back from the cutting edge will its pressure fall upon the tool.

7 Under ideal conditions, the torsional capacity of the drill should be its limit of cutting strength. This does not obtain in the present commonly used types of drill, and they break down very considerably below their torsional strength. Why this is so will be explained later in detail.

8 The feed remaining constant, the horsepower consumption

will be proportional to the speed. This is true both of the power consumed by the machine and that consumed in cutting. The speed remaining constant, the power consumed in cutting does not increase as rapidly as the feed and the power consumed by the machine remains constant for all feeds.

9 From this, it will be seen that the most economical method of chip production is by giving preference to the feeds, rather than the speeds. Power, time and drills will be saved thereby.

COMMONLY USED TYPES OF DRILLS

10 Figs. 1 and 2 illustrate the section scheme of drills now commonly used. While there are many modifications of these,

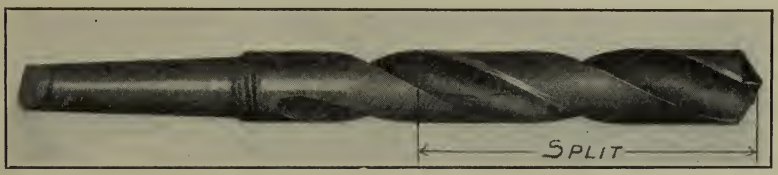


FIG. 4 ORDINARY TYPE OF DRILL SHOWING HOW DRILL SPLITS CENTRALLY IN HEAVY USAGE

the figures suffice to illustrate their common characteristic, which is that the cutting edges A and B pass to one side of the axis of motion of the drill instead of through the axis. In this respect there is no difference between Figs. 1 and 2.

11 It will be seen that in this scheme the drill has four distinct edges, A, B, C and D, and that the usual name given to it of "two-lip" drill is not correctly applied.

12 Referring to Fig. 3, it is customary for the included angle E to be of 118 deg. and the cutting edges C and D, Figs. 1 and 2, therefore have an unfavorable lip angle. These edges constitute what is commonly called the chisel point and their cutting resistance is very great. The cutting edges A and B cut more freely than the cutting edges C and D and a tendency to longitudinal fracture of the drill is set up thereby. This is the cause of the splitting of drills (Fig. 4). In explanation of this splitting tendency, Fig. 5 represents a section of brittle wire A held in vise jaws B. If sufficient pressure is applied, as indicated by arrows X and Y, the wire will break at R and S, due to the resistance

of the vise and to the forces X and Y tending to revolve it about its axis.

13 Those portions of the wire not within the vise jaws represent, in the drill, the cutting edges A and B, Fig. 1, free cutting, due to their favorable lip angle. That portion of the wire held within the jaws represents the cutting edges C and D, or the chisel point, imbedded in the work and having to overcome a high cutting resistance due to the unfavorable cutting lip angle of these edges. Under very heavy feeds, these two edges tend to stand still, while edges A and B continue to cut with the result that the drill is fractured in a manner similar to the breaking of the wire in Fig. 5. In the drill the width of the vise jaws becomes infinitesimal and the fractures R and S coincide and constitute the longitudinal splitting of the drill.

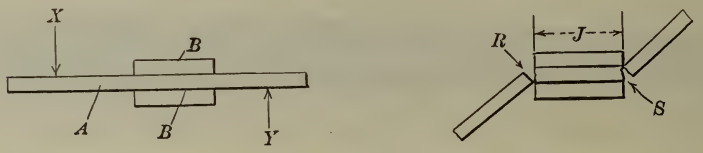


FIG. 5 ILLUSTRATING SPLITTING ACTIONS BY MEANS OF WIRE HELD IN VISE

14 In the commonly used types of drills there are but two methods of reducing the chisel point, viz., by thinning the drill at its center, or by pointing. By the former method the resistance to longitudinal splitting becomes lowered and by the latter method the cutting edges lose their support at and near the center of the drill.

15 By the foregoing it will be seen that the feed possibilities of the ordinary types of drills are not very great. Either the drill will split or its cutting edges will break under heavy feeds, long before its torsional capacity is reached. Summed up, the disadvantages of this type of drill are as follows:

a Its weakness.

b The unfavorable cutting lip angles of the chisel point which create a tendency to longitudinal splitting of the drills.

c If the chisel point is reduced by central thinning of the drill, the resistance to longitudinal splitting is lowered.

d If the chisel point is reduced by pointing, the cutting

edges are deprived of their essential support at and near the center of the drill.

e Essential central thickness and cutting edge support are obtained only in connection with a considerable chisel point.

f The tendency to longitudinal splitting is increased by any increase in the extent of the chisel point.

g The resistance to penetration due to the chisel point is very great and is increased by any increase in the extent of the chisel point.

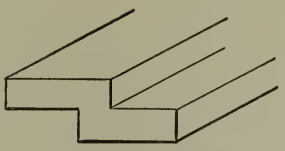


FIG. 6 DIAGRAM ILLUSTRATING PRINCIPLE OF VAUCLAIN DRILL

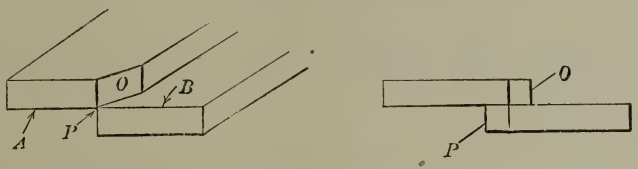


FIG. 7 CENTRAL EDGES OF FIG. 6 SHOWN BEVELED

h It is not adapted to heavy feeds, and must therefore be used under the conditions of moderate feeds and high speeds, the least economical method of chip production.

i There are four cutting edges, two of which do not pass through the axis of motion, hence the aggregate length of the cutting edges is excessive. This is productive of greater torque, hence of power consumption.

DESCRIPTION OF THE VAUCLAIN DRILL

16 The Vauclain drill is a "heavy feed" drill adapted to the most economical method of metal cutting and of great strength, due to its design.

17 In order better to describe the drill, the section scheme will first be explained diagrammatically and the development of the actual section will then be illustrated.

18 In Fig. 6 is shown a bar comprising two flat bars overlapping and integrally connected, as indicated.

191 By beveling, as shown at *O* and *P*, Fig. 7, the edges *A* and *B* are made to meet at the axis of the bar. Since these

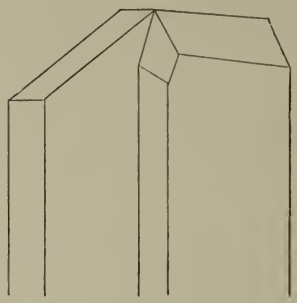


FIG. 8 SHOWING FIG. 7 GROUND TO FORM A DRILL

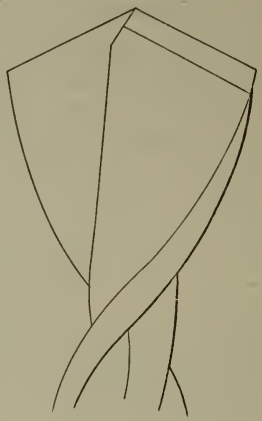


FIG. 9 SHOWING APPEARANCE OF DRILL WHEN TWISTED

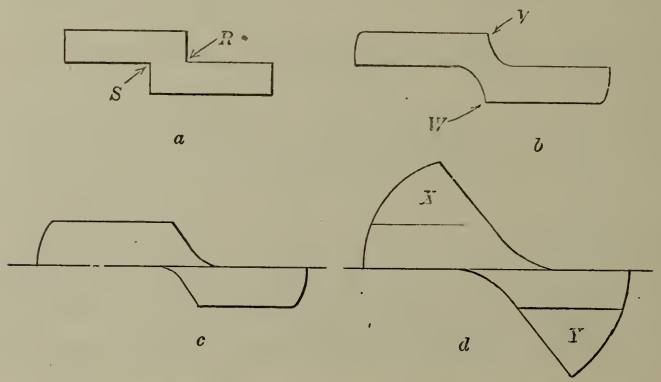


FIG. 10 SUCCESSIVE SECTIONS SHOWING DEVELOPMENT OF BAR OF FIG. 6 INTO FORM ADAPTED FOR VAUCLAIN DRILL

beveled surfaces are at an angle with the axis the integral connection between the two flat bars remains unbroken.

20 Fig. 8 indicates how the end of the bar would appear when ground to form a drill and Fig. 9 how it would appear if twisted.

21 Fig. 10-*a* again shows the section or end view of the bar in its original form. Introducing fillets at *R* and *S* gives the form shown in Fig. 10-*b* and removing the corners *W* and *V* modifies it as in Fig. 10-*c*. In Fig. 10-*d*, by adding areas *X* and *Y* to the section, its torsional value is increased.

22 Fig. 11 shows the exact section profile and in Fig. 12 is reproduced a photograph of three of these drills.

23 Tests have been made for the purpose of comparing the

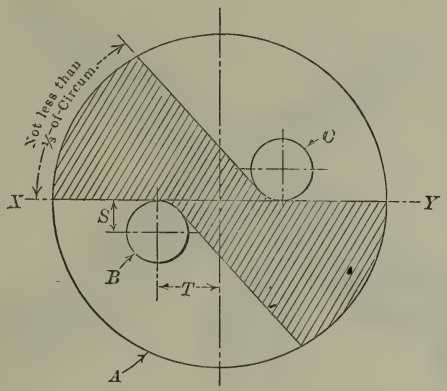


Fig. 11 Diagram showing Exact Form of Construction of Drill

Circle *A* = Circumference of Drill
Diameter of Circle *B* = 3/16 Diameter of Drill
Diameter of Circle *C* = 3/16 Diameter of Drill
S = 3/32 of Diameter of Drill
T = 3/16 of Diameter of Drill
XY = indicates cutting edges

ordinary and the Vauclain types of drills. In order to secure uniform conditions care was taken that the different drill tests should be alike in quality of drill steel, heat treatment and tempering. The material drilled was a tough forging grade steel of about 0.45 carbon. Tables 1, 2 and 3 show the horsepower saved by giving the preference to the feeds, also the relative strength and endurance of the ordinary and the Vauclain types of drills. The size of drill used in all tests was $1.^{9}/_{64}$ inches.

24 By examination of these diagrams and photographs it will be seen that the chisel point is eliminated without central thinning of the drill and without weakening the cutting edges.

ADVANTAGES OF THE VAUCLAIN DRILL

25 The improved design of this drill has brought about the following important features:

a Essential central thickness is secured.

b Cutting edges are properly supported at all points.

c Chisel point, with its unfavorable lip angles, is eliminated without weakening drill or cutting edges.

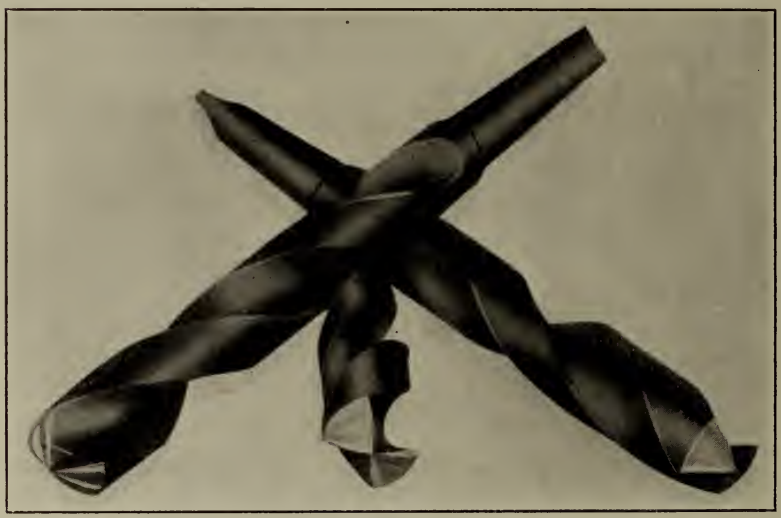


FIG. 12 EXAMPLES OF THE VAUCLAIN DRILL

d Cutting edges pass through axis of motion.

e There are only two cutting edges.

f Aggregate length of cutting edges is reduced to minimum.

26 As a result of these features the following advantages are secured:

g Increased strength and reduced liability to splitting.

h It is adapted to heavy feeds, the most economical method of cutting metal.

i Productive capacity is increased.

j Horsepower consumption per unit of metal removed is reduced.

k Life of drill is increased and frequency of grinding is less.

TABLE 1 STRENGTH COMPARISONS

Test	Kind of Drill	Feed per Rev.	Speed R.p.m.	Inches per Min.	Holes Drilled	Total Inches Drilled	Compare to Test	Condition of Drill After Test	See Note
1	Regular	0.009	300	2.70	1	3.875	6	O. K.	
2	Regular	0.0145	300	4.35	1	3.875	7	O. K.	
3	Regular	0.020	300	6.00	1	3.875	8	O. K.	*A*
4	Regular	0.030	300	9.00	1	3.875	9	O. K.	
5	Regular	0.050	300	15.00	0	0. —	10	Broke—see note	
6	Vauclain	0.009	300	2.70	1	3.875	1	O. K.	
7	Vauclain	0.0145	300	4.35	1	3.875	2	O. K.	
8	Vauclain	0.020	300	6.00	1	3.875	3	O. K.	*B*
9	Vauclain	0.030	300	9.00	1	3.875	4	O. K.	
10	Vauclain	0.050	300	15.00	1	3.875	5	O. K.—see note	
11	Regular	0.009	170	1.53	1	3.875	16	O. K.	
12	Regular	0.0145	170	2.465	1	3.875	17	O. K.	
13	Regular	0.020	170	3.40	1	3.875	18	O. K.	*C*
14	Regular	0.030	170	5.10	1	3.875	19	O. K.	
15	Regular	0.050	170	8.50	see note	1.000	20 note	Broke—see note	
16	Vauclain	0.009	170	1.53	1	3.875	11	O. K.	
17	Vauclain	0.145	170	2.465	1	3.875	12	O. K.	
18	Vauclain	0.020	170	3.40	1	3.875	13	O. K.	*D*
19	Vauclain	0.030	170	5.10	1	3.875	14	O. K.	
20	Vauclain	0.050	170	8.50	1	3.875	15	O. K.—see note	
21	Regular	0.009	170	1.53	1	3.875	26	O. K.	
22	Regular	0.145	170	2.465	1	3.875	27	O. K.	
23	Regular	0.020	170	3.40	1	3.875	28	O. K.	*E*
24	Regular	0.030	170	5.10	see	3.875	29	Broke emerging	
25	Regular	0.050	170	8.50	note	.....	30-31-32	.....	
26	Vauclain	0.009	170	1.53	1		21	O. K.	
27	Vauclain	0.0145	170	2.465	1		22	O. K.	
28	Vauclain	0.020	170	3.40	1		23	O. K.	
29	Vauclain	0.030	170	5.10	1	see	24	O. K.	*F*
30	Vauclain	0.050	170	8.50	1	note	25	O. K.	
31	Vauclain	0.050	208	10.40	1		25	O. K.	
32	Vauclain	0.050	245	12.25	see note	1.000	25	Broke—see note	

A Drill broke after point had penetrated to 7/8 in. diameter, broke into many pieces; split. Torque is practically proportional to the square of the diameter, hence, since *A* broke when point had penetrated to only 7/8 in. diameter, *B* showed 170 per cent the strength of *A*.

B Drill was permitted to emerge from work without knocking out feed; a slight crack in one cutting edge developed after drilling at rate of 15 in. per min. and emerging.

C Drill broke into many pieces after penetrating 1 in. in last hole. Drilling at rate of 8.50 in. per min. drill split.

D Since no heavier feeds were obtainable in the machine this drill was afterward run at 208 r.p.m. and 0.050 feed and broke only when emerging from bottom of hole; drilling at rate of 10.40 in. per min. and emerging. Taking into consideration that *D* emerged and *C* did not, the difference in strength is much greater than that indicated by the difference in inches per minute.

E Split longitudinally.

F In order to get an actual comparison with *C* which broke at 1 inch penetration, this drill was not allowed to emerge from the work. It broke drilling at rate of 12.25 in. per min.; after penetrating 1 in. Drill broke torsionally and did not split.

l Distress in machine is less and resistance to penetration is reduced.

m Better holes are produced and afterwork of reaming is less.

TABLE 2 ENDURANCE TESTS

Test	Kind of Drill	Feed per Rev.	Speed R.p.m.	Inches per Min.	Holes Drilled	Total Inches Drilled	Compare to Test	Condition of Drill After Test	See Note
33	Regular	0.009	300	2.70	42	162.75	34	Cutting edges worn	*A*
34	Vauclain	0.020	208	4.16	42	162.75	33	Cutting edges worn	*A*
35	Regular	0.020	300	6.00	21	81.37	36	Cutting edges burned	*B*
36	Vauclain	0.030	208	6.24	21	81.37	35	Cutting edges O.K.	
37	Regular	0.020	300	6.00	6	23.25	38-39	Badly burned	
38	Vauclain	0.030	208	6.24	26	105.75	37	Cutting edges O.K.	*C*
39	Vauclain	0.030	208	6.24	35	135.12	37	See note	*D*
40	Vauclain	0.0145	208	3.02	20	...	41	O. K.	
41	Regular	0.009	208	1.87	20	...	40	O. K.	
42	Regular	0.009	300	2.70	11	...	43	O. K.	
43	Vauclain	0.0145	245	3.55	11	...	42	O. K.	
44	Regular	0.009	300	2.70	10	...	45	O. K.	
45	Vauclain	0.0145	245	3.55	10	...	44	O. K.	
46	Vauclain	0.020	245	4.90	5	...	47	O. K.	
47	Regular	0.0145	245	3.55	5	...	46	O. K.	
48	Regular	0.0145	300	4.35	10	...	49-52	O. K.	
49	Vauclain	0.020	300	6.00	10	...	48	O. K.	
50	Regular	0.020	300	6.00	25	...	51	Burned corners	*B*
51	Vauclain	0.020	300	6.00	39	...	50	Burned corners	*E*
52	Vauclain	0.020	300	6.00	10	...	48	O. K.	

A Cutting edges of Vauclain drill *less* worn than regular drill.

B Chisel point also burned.

C Broke in 26th hole by running into another hole.

D Spindle of machine ran hot, causing "jumping" of speed, and cutting edges; broken by resulting shocks. Drill at one period stood still, but did not fracture except at cutting edges.

E Point O. K.

TABLE 3 COMPARATIVE TESTS OF REGULAR AND VAUCLAIN DRILLS

Test	Kind	Feed per Rev.	Speed, R.P.M.	In. per Min.	H.P. Expended	Per Cent Saving
53	Regular	0.00599	300	1.797	10.86	....
54	Regular	0.02	87	1.74	10.30	....
55	Vauclain	0.02	87	1.74	5.03	51.2
56	Vauclain	0.00599	300	1.797	4.73	56.4

DISCUSSION

J. Sellers Bancroft asked if any efforts have been made to ascertain the accuracy of diameter of holes drilled with the Vauclain drill. He thought it would be difficult to get the point of the drill exactly central, and in the production of holes of accurate size it is necessary for the reamer, which follows the drill, to have a definite amount to take off throughout. This amount should be neither more nor less.

H. P. Fairfield. The report of the horsepower expended as given in Table 3 shows a remarkable difference between the regular and the Vauclain drills. It would seem as if such differences in results could not be due to the design of the drills themselves, but must be the result of the method of testing. It would be interesting, therefore, to know how these tests were made and the apparatus used to measure the efficiency of the several drills. The reader could then analyze the tests and judge for himself as to their reliability and exactness. In much of the ordinary drill testing, the weaknesses and friction losses of the drilling machines are registered and charged to the drills being tested, which is obviously unfair. Several years ago at the Worcester Polytechnic Institute a drill testing dynamometer was developed which obviates such errors, registering only what is taking place at the drill point and entirely eliminating the drilling machine from the readings taken. If anyone compares his dynamometer results with the power input of this machine he will be shown conclusively that tests based upon the differences in power input are extremely unreliable. It is for this reason that the writer believes the method of conducting the tests should have been included. In a series of dynamometer tests upon a regular and a Vauclain drill made since reading the paper, the results showed no material differences in horsepower expended, and the writer cannot see why there should be such differences as shown in Table 3. As furnished him, the Vauclain drill is essentially a point thinned drill. Therefore the pressure at the point is reduced and the distortion in the machine lessened. However, if the regular drill is point thinned as is often done, the results in thrust are essentially the same.

The Vauclain drill appears to lend itself more readily to the point thinning principle than does the regular drill, and if ex-

tended use and tests should prove this to be so, it is obvious that an advance in drill construction has been made.

E. C. Peck. This paper points out a great many weaknesses of the well-known twist drill and has attempted to prove the superiority of a new type, but the author has described neither the apparatus nor the exact conditions under which the tests were made, in a way such that the student may judge to what particular detail the difference in results is traceable.

In Par. 1 he says the best definition of economical drilling is rapid drilling; this is not necessarily true. The writer has drilled cast iron 4 in. thick with a 1½-in. drill at the rate of 64 in. per min., probably the record at this date for rapid drilling, but it was not economical because it consumed too much power and the drill was too near breaking all the time. A better definition of economical drilling is "the greatest amount of drilling that can be done per $1, counting the cost of time, power, machinery and drills."

In Par. 8 he says "The speed remaining constant—the power consumed by the *machine* remains constant for all feeds." It would be interesting to know how he determined this, as the coarser the feed the heavier the pressure on the drill to make it penetrate, and this puts more pressure per square inch on thrust bearings and feed parts which in turn increases friction.

Referring to Fig. 1, the cutting edges A and B being ahead of the center presents an advantage that the author has not mentioned, viz., they make a shearing cut with a tendency to carry the curling chips away from the axis of the drill.

The author lays stress on the splitting of drills and demonstrates the reason for this failure, but the splitting of drills is not a serious matter and a split drill usually shows a defect in the material or improper grinding, a fact which any large user of drills can easily verify by careful observation. The writer is in a position to observe the performance of several thousand drills each month which are tested to destruction under severe conditions and not one in 10,000 fail by splitting. If the chisel point spoken of has an inclination to the cutting edges A and B as shown in Fig. 1, there will be enough clearance at the center and the chisel point will cut easily enough to avoid splitting.

A great defect in the author's proposed drill, as shown by the writer's experience, is its inability to penetrate hard material.

This is apparently due to the extremely thin center not being able to withstand the heavy feed pressure necessary to penetrate hard material. A more serious defect is the actual pressure of the chip against the surface *O* in Fig. 7. It does not curl freely as in the drill shown in Fig. 1 because there is no room; it must slide against the shoulder *O*, Fig. 7.

In the regular type drill the cutting edge is extended past the axis an amount which allows free curl to the chips. If the regular and proposed drill used in the author's tests was properly thinned to the same thickness, there should be less power consumed by the regular drill because it bends the chips better.

Tests with Vauclain and regular drills made from the same bar of steel and heat treated alike showed in every case that to withstand heavy feeds (0.100 in.) the temper of the Vauclain drill had to be lowered 20 to 30 points below the regular drills to keep the centers from breaking and they were then too soft for high speeds.

The tables do not show the "rapid drilling" which might be expected from the author's statements. The highest rate given is 15 in. per min. and double this amount can be done with regular drills. It would therefore seem that more extensive tests will be needed to make the superiority of the Vauclain drill apparent.

THE AUTHOR. In reply to Mr. Bancroft's question, accurate grinding of any drill is essential to the production of holes of accurate size. There is no difficulty in obtaining correct grinding in the Vauclain drill. No drill should be hand ground, and proper grinding machinery has been designed and built for grinding the Vauclain drill correctly. It is interesting, however, to note in this connection that much of the grinding of the Vauclain drills during the development was done by hand until the time when proper grinding machinery was available.

Replying to Mr. Fairfield, the author wishes to state that the remarkable difference in horsepower expended was not due to the method of testing but to the comparison between high speeds and heavy feeds. In other words, with a given rapidity of drilling a high-speed drill is made to depend upon high speeds and moderate feeds. The Vauclain drill utilizes heavy feeds and moderate speeds, the speed being kept just high enough to main-

tain within the drill a sufficiently high temperature to operate most successfully.

The author is not aware of any Vauclain drills having been furnished the Worcester Polytechnic Institute for the purpose of tests, and is of the opinion that perhaps such tests may have been made without a proper understanding of the intent of the drill design.

In reply to Mr. Peck, the apparatus and conditions under which the tests of the Vauclain drill were made were as follows:

The drilling machine used was a 60-in. radial drill press, one of a number built by Wm. Sellers & Co., Inc., of Philadelphia, and driven by a 20-h.p. variable speed direct-current motor. Current was supplied by an independent generator so that the machine was entirely segregated from shop lines, the object being to maintain as constant a voltage as possible. A Westinghouse recording wattmeter, also an ammeter and voltmeter for direct readings, were located about 75 ft. from the machine so that readings could be taken without encountering the dangers of flying parts of broken drills. In this same location there was installed a cut-out switch so that the machine could be stopped at any time without approaching it. A steel guard was placed about the drill spindle as a further protection against flying pieces of broken drills.

The material drilled was tough, hammered steel of approximately 0.45 carbon. Heavy slabs of this material were used so that holes would have to be drilled through a considerable depth in all cases.

The author in stating that the best definition of economical drilling is rapid drilling believes he is correct and that Mr. Peck has misunderstood the statement. The most costly element in drilling is that of time; and, by the saving of time, assuming a rational design of drill and of horsepower consumption, the amount of drilling that can be done per unit of cost will increase as the time consumed decreases.

In Par. 8, referring to the statement, "the speed remaining constant, the power consumed by the machine remains constant for all feeds," the author should perhaps have explained this matter more in detail. It is true, as Mr. Peck has stated, that the coarser the feed the heavier the pressure on the drill to make the cut, and this puts more pressure on the thrust bearings and feed parts, which in turn increases friction. Attention is called to

the fact, however, that the friction due to end thrust of the drill is so small as compared with the total power consumed by the friction of the machine that it may be considered negligible so far as power consumption is concerned. That the power consumption varies directly as the speed is but a mechanical law which may be verified at any time and by anyone by electrical readings. To elaborate this further, assume that with a certain size drill and a given rate of feed and speed there is a gross consumption of 6 h.p., and that half of this power is consumed by the machine, leaving a net cutting horsepower consumed of 3. Next assume an approximate doubling of the rapidity of drilling. If this is obtained by doubling the speed, there will be a gross consumption of approximately 12 h.p. If, however, the rapidity of drilling is approximately doubled by increasing the feed and permitting the speed to remain unchanged, the gross consumption will be approximately somewhat less than 9 h.p. In a twist drill of the ordinary design, the difficulty of penetration and consequently the end thrust of the drill is aggravated by the chisel point. In the Vauclain drill, there being no chisel point, the end thrust upon the drill is very much less, consequently heavier feeds can be used in the Vauclain drill with less distress to the feed mechanism than would occur with the use of the ordinary type of twist drill.

The author cannot agree with Mr. Peck's statement that the splitting of drills is not a serious matter, and that the drill usually shows a defect in the drill material or improper grinding. The author has had a very large and intimate acquaintance with drills of all kinds. The central weakness of a milled drill is well known to all drill users. In the break-down tests made for the purpose of comparing the Vauclain drill with the ordinary design of drill, the ordinary designs of drill almost invariably split longitudinally, while the Vauclain drills, standing up under very much heavier duty as shown by the test data in the paper, fractured only when the torsional capacity of the section had been reached. Some of the drill manufacturers today, appreciating the seriousness of the central longitudinal weakness of the ordinary drill, have thickened their drills in the center, but this has increased the chisel point, consequently the difficulty of penetration. In the Vauclain drill there is no chisel point, consequently it may be made sufficiently thick in the center to enable the full torsional value of the drill section to be realized.

The author must disagree with Mr. Peck's statement in reference to the inability of the drill to penetrate hard metal. The Vauclain drill is especially adapted to the penetration of hard metal. It has no "extremely thin center." The so-called point on the Vauclain drill is merely the absence of a chisel point and is the strongest and most lasting and perfect part of the drill. It will not burn and it will not break under duties which would destroy the drill in every other respect.

With regard to the chip not curling, this is also incorrect. Since the cutting edges pass through the axis of motion, the chip at its inner edge has a length of zero while the maximum length of the chip is at the outer edge, consequently the outer edge of the chip is in rapid upward motion while the inner edge is approximately at rest. These circumstances, combined with the flute of the drill, give the chip the proper curl. No difficulty has been experienced with the Vauclain drill in this respect.

As to the temper of the drills having to be lower than that of the regular drills, this is not at all necessary, as stated by Mr. Peck, to prevent the centers from breaking. The author has stated that the Vauclain drill is a heavy-feed drill and has pointed out and substantiated the fact that high-speed drilling is less economical than heavy-feed drilling, also that high-speed drilling is the result not of desire or advisability, but rather of inability to obtain, in the ordinary type of drill, any great amount of physical strength. In the Vauclain drill a tough rather than a hard drill is required, and it is intended that these drills shall be drawn at a lower temperature than the high-speed drill of ordinary type.

With reference to the drilling of 15 in. per minute, it is to be understood that this was done in a piece of 0.45 carbon hammered steel, approximately 4 in. thick and that the drill was permitted to emerge from the bottom of the work under full feed without the slightest injury to the drill. In the author's opinion, the drilling of 64 in. per minute with 1½-in. drill in cast iron is not to be compared in severity with the test just referred to. The greatest evil of end thrust on an ordinary drill is that under heavy duty it is apt to break when emerging from the work. With the Vauclain drill feeds of ⅛ in. per revolution have been successfully carried with a $1\,^{9}/_{64}$-in. drill at high speeds.

No. 1369

EFFICIENT PRODUCTION OF CYLINDRICAL WORK

By C. H. Norton,[1] Worcester, Mass.

Non-Member

Turning is one of the oldest of the mechanic arts, and the lathe is one of the oldest metal working machines; but the lathe has been a very inefficient tool until the present time. We now have lathes so designed that they can be made efficient. I believe, however, that we are not as a rule using the modern high power lathes as we should, and are losing what they were designed to save, viz., *time;* that while we use them as roughing machines we also use them as refining machines.

2 Most cylindrical work must finally have size within small limits and be truly cylindrical within still smaller limits, which led mechanics in the past to study the art of refined turning. When this art had reached a high degree of refinement, men discovered that it did not satisfy them and cylindrical grinding was introduced to give more refinement than could be obtained with the lathe. The natural conclusion at that time was that the grinding machine should take up the work of refinement after the limit of refinement had been reached with the lathe and the early grinding machines were, therefore, designed with this thought in view.

3 Later, however, it was shown that ground cylindrical work could be produced at a cost no greater than the turning alone had cost, simply by using the lathe to remove a large proportion of the metal necessary to be removed and not at all as a means of refinement; transferring the entire responsibility for refinement to the grinding machine.

4 While some understand this and are producing cylindrical work efficiently, I believe the great majority are still under the

[1] Norton Grinding Company

Presented at the Annual Meeting 1912, of The American Society of Mechanical Engineers.

spell of tradition, and are turning what they call "good work," work that shows care and skill on the part of the operator and accuracy in the lathe. Such care and skill with turned work is even more senseless today than the old time practice of shaping acorns on all screw heads and putting cast-iron eagles on the tops of machine tools, because in case of the former the grinding machine removes all trace of the lathe's accuracy and the operator's skill in a moment of time, while the latter remained for years to gratify the taste of those who had an eye for such artistic embellishment.

5 The developments of the last few years have brought about the facts that, except in rare cases, efficiency in the production of cylindrical work means, (*a*) the use of the lathe as a roughing tool only, and (*b*) the modern grinding machine for refinement.

6 Most lathe operators and most foremen seem to understand roughing to be simply the turning of work a few thousandths over the finish size, when in reality they are not roughing at all, but simply turning to a certain degree of refinement with the notion that by so doing they are (*a*) doing a creditable job, and (*b*) that they are helping the grinding machine. It is difficult to secure roughed work from lathe operators, i.e., work that is not in any sense refined.

7 Tradition impels nearly all to consume much more time than would be necessary if the work were merely roughed, roughing being confused with a certain degree of refinement. Roughing is not refinement of any degree, and refinement does not necessarily mean perfection. There are many degrees of refinement, but never degrees of accuracy or perfection. Accuracy and perfection are final and fixed, they have no limits.

8 Those who would produce cylindrical work efficiently must recognize the fact that different cases require different degrees of refinement, and whether the work requires a very low degree of refinement or a degree closely approaching accuracy, the lathe work, to be efficient, should be the same, viz. as *rough* and as *cheap* as it is possible to make it.

9 The lathe today should be a roughing tool only, and the grinding machine a refining tool only, not a perfecting tool. Some grinding machines may produce greater refinement than others, but none can produce literally exact and perfect cylinders. While it is true many mechanics look upon a high degree of

refinement as accuracy and speak of it as such, it is easy to prove that it is not accuracy, but simply a high degree of refinement. When all recognize these facts there will be greater efficiency in the production of cylindrical work.

10 Because these facts about roughing and refining are not well understood and men do not combine the lathe and grinding machine in a well defined and efficient manner, the machine industry and railroads are losing large sums each year and I will try to show where some of the loss occurs.

11 We are losing because we turn closer than *about* 1/32 in. above the finish size. I use the word *about* because we lose by turning carefully. To require lathe work turned to thousandths today is like burning money for the fun of it. We are losing by allowing workmen to use any but the coarsest feeds possible in each case, even to the use of the screw-cutting gears to obtain them.

12 Feeds as coarse as four to the inch should be used in some cases and six, eight and ten should be quite common. When work is too frail to allow such feeds with one cut, then another, or even three cuts should be made at faster cutting speed; for, while it is true that very coarse and imperfect turning will require longer to grind than close, careful turning, the combination of coarse, cheap turning with grinding will produce cylindrical work more efficiently. The modern grinding machine will remove metal most efficiently when that metal is in the form of coarse screw threads.

13 High ridges do not increase the cost of production. We lose money when workmen are allowed to caliper cylindrical work on the tops of the ridges, as nearly all do. This accounts for the misunderstanding about coarse turning and the allowance for grinding. Calipering on the tops of the ridges makes it necessary to turn with relatively fine feed to enable clean grinding. To avoid this some workmen use a broad nose tool with the coarse feed which necessitates slower cutting speed because more metal is removed than with a pointed or grooved cutting tool. Ridges left by a more pointed tool do not increase the cost of producing that work, but decrease it.

14 It is so simple to feed by hand a slight distance at the end of the work, where the calipering can be done correctly, however coarse may be the power feed later, and however high

the ridges. Money is lost by allowing workmen to measure roughed work elsewhere than at the end; hours are wasted by operators calipering at several points and then because of spring or tool wear, resetting and turning that portion again to secure straight work. They forget that the modern grinding machine was designed to do all of the straightening and all of the sizing, regardless of what the errors of roughing may be. They forget, or have never known, that the art of producing cylindrical work has progressed beyond that stage where it was necessary for the lathe to do its best before the grinding machine assumed the work of refinement.

15 The modern grinding machine is literally a correcting machine. The roughing can literally be rough, and roughness does not mean an approximate refinement. It is easy to figure the saving in time of turning a piece of work 6 in. in diameter by 6 ft. long when the last cut before grinding is made with a feed of six per inch, instead of 32 per inch which is very common. When it is considered that the grinding machine will grind this six per inch work to the finish size complete in less time than is required to file 32 per inch work, we see the great possibilities for saving cost with the lathe, if combined with a modern grinding machine.

16 In some cases money is lost by roughing at all in the lathe. Please note that I say *in some cases*. Efficient production of cylindrical work is usually accomplished by roughing first in the lathe; but there are cases when the difference in favor of grinding without first roughing in the lathe is very great.

17 The most important study for those who would secure efficient production of cylindrical work is the roughing of that work. It is here that the greatest possibilities for saving lie. If all would make a careful study of the roughing preparatory to grinding instead of trying to improve upon the methods of grinding advocated by grinding machine makers, there would result a more efficient production of cylindrical work and a clearer view of the real reasons why the grinding machine maker advocates certain methods, and why the modern heavy grinding machine was introduced.

18 At a works where large numbers of cylindrical pieces are manufactured 1⅛ in. in diameter and about 10 to 11 in. long, 5 minutes was required to turn each, removing about 1/16 in. from

the diameter using a high-speed steel tool and revolving as rapidly as the tool would stand. The plan was to turn close to the finish diameter and care was taken to secure straight, smooth and round work to save time when grinding, because they considered the grinding expensive, and that this expense must of course be added to the cost of turning. The grinding in this case was 1 minute. The total time for producing these pieces was 6 minutes.

19 A change was made in the shape of the tool point and the size limit for turning increased. The traverse feed was increased, the cutting speed remained unchanged and the work was turned in 1 minute each, while the grinding time was doubled, viz., 2 minutes; the saving was 3 minutes, or one-half of the original time.

20 Some idea of what is lost by allowing lathe work to be turned straight, smooth and close to size before grinding may be obtained from the following illustration of actual work on some forgings about 4 ft. 6 in. long, turned portion to finish 2 in. plus or minus 0.0005 in. in diameter, and 3 ft. long. The lathe operator turned these in the same manner that the majority of operators are now turning such work, viz., as such work was turned before grinding machines were introduced, except that 0.010 in. was left on the diameter for grinding. The turning required two cuts, as there was more than ⅛ in. to remove and the work was somewhat frail as well as irregular. The first cut was considered by the workman a very coarse heavy cut; the second was like the last cut in the majority of shops, viz., it gave what the lathe men and most foremen call "a good job." When ready for grinding the time consumed by turning was a total of 25½ minutes. This appears to be too slow, but the forging was too slim to allow little, if any more, with one rough cut, and the lathe was an old-time weak power frail machine. The grinding time on this was 3 minutes, making a total of 28½ minutes.

21 When tradition was ignored and the work was made ready for grinding in the proper way, the following saving was accomplished: Two roughing cuts were taken, because (*a*) the lathe was not powerful, and (*b*) the work was frail. The feed was 10 per inch, cutting deep grooves; the time complete ready for grinding was 9 minutes. The work was not straight, and was what lathe men would style "a very poor job." The grind-

ing time to the exact limits was 9 minutes, or three times longer than with the more careful turning; a total of 18 minutes, a saving of 10½ minutes on each forging.

22 An attempt to rough this work with one cut instead of two resulted in a loss of 2 minutes. A modern lathe having more power with the direct belt would have carried the same feed and depth of cut at higher cutting speed with a still further reduction of time. This shows what may be accomplished with weak low power lathes.

23 To install high power lathes and continue to turn just as smoothly and accurately as of old, simply allowing a few thousandths for grinding, would seem the height of wastefulness. Whether high power lathes or old low power lathes are used mechanical industries are losing an enormous sum each year because of the lack of understanding of the real problem affecting the efficient production of cylindrical work, which is the proper combination in each individual case of rough turning and grinding.

24 Some idea of what we are losing by not wearing our "thinking caps," is shown by the following: Some plain shafts, 1 5/16 in. in diameter, 62 in. long were to be made to ordinary limits of size and finish. Tests were made to determine whether it was better first to turn these or to grind direct from the rough bars. The turning required 6 minutes ready for finish grinding, while to grind off the same amount as was turned off required 9 minutes. A number of mechanics pronounced it cheaper to turn the shafts first. "Of course." But in this case after turning in 6 minutes, each shaft required straightening before grinding, which consumed 10 minutes, totaling 16 minutes. When the roughing was done by grinding instead of turning no straightening was necessary, and the roughing cut was removed in 9 minutes.

25 In this case, therefore, the turning was actually lost. There may be similar cases where the shafts will require some little straightening even if roughed by grinding, so that each case should be investigated instead of making a rule that all must or must not be turned first. Usually, however, no straightening is necessary when the roughing is done by grinding.

26 Money is being lost because the majority of designers have not kept pace with the development along these lines. A

well-known runabout motor car has a crankshaft, which if made about 1 in. longer over-all could be produced for 50 cents less on each crank. When it is considered that about 30,000 of these cranks have been made, representing a possible saving of $15,000, we see the importance of considering the combination of lathe and grinding work.

27 Fig. 1 shows a design of crankshaft that is expensive to machine, since it requires carefully finished lathe work. It will be seen that this shaft has a flange that must be finished complete with the lathe. Aside from the flange also the design necessitates lathe finishing of the majority of the work. Grinding is a small part of the whole.

28 Fig. 2 shows a design that requires no lathe work whatever, there being clearance for the sides of the grinding wheel at all points. The illustration shows the six operations necessary to complete the shaft after cutting off the ends and centering.

29 These shafts are finished to the usual limits directly from the drop forging by grinding. The net labor cost for cutting off ends, centering, and machining complete with one keyway and the thread ready for the automobile, is 55 cents. The cost for grinding wheels is 2½ cents each; cost for diamonds to true and shape the wheel, is 2 cents each.

30 Another feature affecting the cost of production is the matter of ordinary shoulders. This is overlooked by many designers and practically all draftsmen. It is easy to draw two sharp lines that cross each other and form a sharp corner and the fact that it costs less to provide room for the shoulder fillet on the other member than to make a sharp corner in the cylindrical member is lost sight of. This no doubt is the result of the experience of the past when there were no powerful grinding machines to locate and form such shoulders cheaper than the lathe. Shoulders to be made with the grinding wheel require a slight fillet.

31 When numbers of duplicate pieces are to be produced and a small fillet is allowed, not only can the cylindrical portion be roughed with the lathe and not at all refined, but the shoulders can also be roughed, because the grinding machine, when required, is provided with a locating bar. By the use of this bar, the exact location of all shoulders can be secured by grinding and without measuring. At the same time the cylindrical

portion can be ground with the same grinding wheel. The roughing in the lathe can be done with the same tool that roughs the cylindrical portion because the lathe in this case is not required to give to the shoulder any particular shape. When it is found best to *locate* and *finish* shoulders with the lathe because

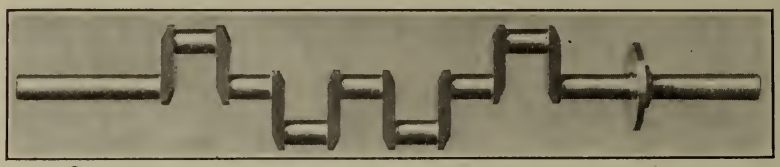
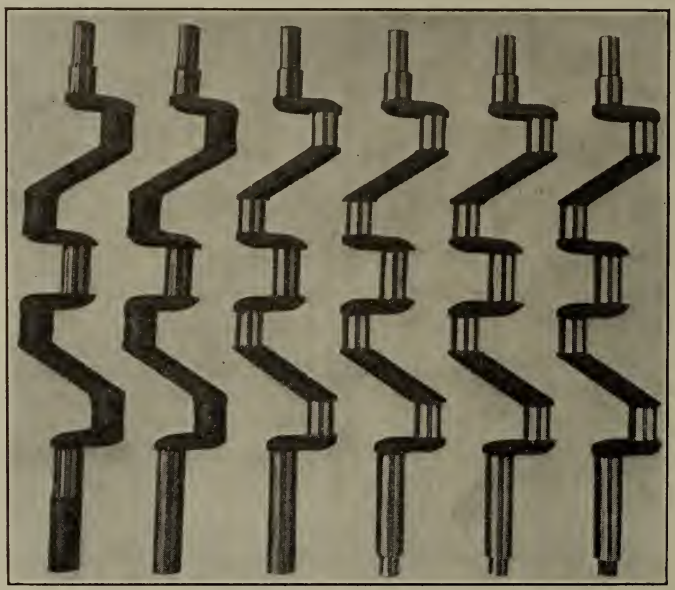


FIG. 1 EXAMPLE OF EXPENSIVE DESIGN



1 2 3 4 5 6

FIG. 2 EXAMPLES OF INEXPENSIVE DESIGN

there is no locating bar for the grinding machine, it should not be necessary to neck in at the shoulders for grinding. This is an old and out-of-date notion, for, with a shoulder cut sharp by the lathe tool, the grinding does not materially change the corner.

32 A more efficient method when no locating bar is at hand

is to rough the work in the lathe, leaving the shoulders about 1/32 in. long and whatever angle the roughing turning tool may form. Next grind the cylindrical portion and at the same time cut out the angle at the shoulder with the grinding wheel. The

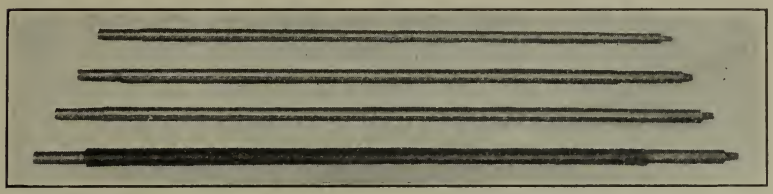


FIG. 3 WORK TOO SLIM TO TURN BEFORE GRINDING

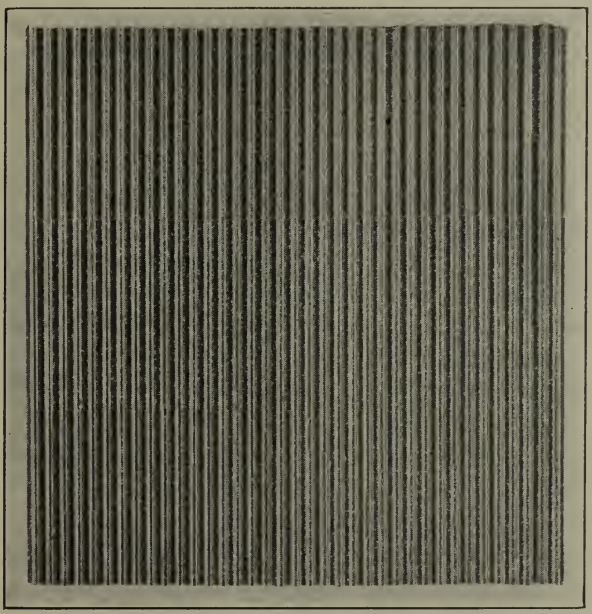


FIG. 4 WORK GROUND WITHOUT TURNING

work should then go to a lathe where a more skilful operator can locate the shoulders exactly; if a sharp corner is really necessary it can then be made.

33 Fig. 3 shows work so slim that turning of the main portion is out of the question if it is to be produced efficiently. The

main or center portion should be ground direct without turning.

34 Fig. 4 shows 37 slim shafts ground direct from the black stock. To turn them first would be a waste of time. They were 30 in. long, 11/16 in. in diameter in the rough, ground to 0.625 in. plus or minus 0.0005 in.; handled twice and ground complete in 17 minutes each. These were tool steels.

35 A lot of 60 countershafts, 35 point carbon steel, finished size 2 3/16 in. plus or minus 0.0005 in. by 66 in. long, turned 8 pitch, leaving about 1/32 in. for grinding; ground complete at the rate of 35 minutes each; cubic inches ground off, 278; cubic inches of steel removed for every cubic inch of wheel wear, 20.7; total cost of wheel for 60 shafts, 38 cents; total cost of power, 35 cents; wheel used, 24 combination grade L alundum; radial depth of cut, when roughing, 0.001 in.; radial depth of cut, when finishing, 0.0005 in.; surface speed of work, roughing, 42 ft. per min.; surface speed of work, finishing, 35 ft. per min.; table traverse, roughing, 11 ft. per min.; table traverse, finishing, 10 ft. per min.; steady-rests "multiple system."

36 To produce cylindrical work efficiently we must make a study of all that affects the problem in each individual case instead of following tradition which is costing us thousands of dollars each year. Lathe builders are offering some very efficient roughing lathes, but are we roughing with them? I think not in most cases. The real roughing lathe needs a good system of multiple steadyrests to enable rapid and deep coarse cuts. Will someone come forward with such a system of steadyrests?

37 Instead of our lathe departments and grinding departments being separate as is common now, there should be one department of cylindrical work under one head. This would check the tendency of the lathe department to treat their work as a finished product, and the tendency of the grinding department to insist upon close careful turning in order to make a better showing of the grinding time, regardless of the real cost of production.

38 There is much yet to be learned about the preparation for grinding of cylindrical work; no two pieces of work require the same lathe treatment. There is a great need for thinking men as foremen and operators. Such men can effect great savings by working out in each case the best combination of turning and grinding of cylindrical work.

DISCUSSION

R. H. Rice said that in his opinion one of the most important suggestions made in the paper is that the lathe and grinding departments should be one and the same. As soon as this idea is well inculcated in our machine shops, the full benefit of the point which Mr. Norton has suggested will be secured.

William Kent stated that efficiency engineers are asking for tabulated data on matters of this kind. They would like, for example, to know the time required for turning and grinding a cylinder of a given size and under given conditions. Such information is too valuable to be hidden in the text of papers and he suggested that where possible an appendix be added to such papers, giving information in tabular form, so that they could be compared with data of the same kind appearing in other papers.

H. P. Fairfield. Mr. Norton's paper is of peculiar interest to me because of a series of tests recently made under my direct supervision in the interests of the Worcester Polytechnic Institute.

The tests were made in a Worcester shop and the aim was to determine in what manner and to what extent the lathe could best be used as a "preparing" machine for cylindrical work finished in the grinding machine. Mr. Norton's remark made in my presence, that a piece of cylindrical work properly prepared for effective grinding should "look like a storm at sea" led to making the tests. The work selected was a lot of forgings to be finished as spindles for heavy machine tools, also a lot of comparatively long and slender shaftings. The lathe and the grinding machine were in these tests considered as a producing unit and worked as such. The first cost of each machine was noted, the power necessary to drive each machine measured and rated, the price paid the operator was charged to each machine and tabulated in the results as was the cost and wear of the cutting tools. The work had the usual amount of excess metal to be removed and the problem assumed the form of a direct question, what is the method of least final cost for removing the excess of stock and what part shall each machine assume in doing the work.

In the case of the forgings the results showed that a certain excess of stock could be removed in the engine lathe with a lesser final cost, using the lathe purely as a roughing out machine, taking coarse feeds and whatever cutting speeds the cutting tools would stand without excessively rapid dulling of the cutting edges. In the case of the longer and more slender shafts it was in some cases found that they could be roughed out and finished at a lesser final cost in the grinding machine without previous lathe operations.

The conclusions drawn were that the cylindrical work in most shops could be well classified into a limited number of groups of machine parts arranged according to their weight, form and material. Also, that the lathes and grinding machines should be arranged as a production unit and treated in all cases as such. Further that each class of parts should receive an accurate time and cost study with the results carefully tabulated and analyzed; that time and cost studies should be frequently made to determine whether or not the classification of parts as arranged was the better one and whether the machines and tools used were the best obtainable for the object sought.

The Author. The facts brought out in this paper are the result of 12 years of patient work and experimenting. The majority of metal machine makers still labor under the impression that grinding is an additional operation in the creation of cylindrical work, and that they have simply to turn work as they have always done, allowing something for the grinding machine to grind off. That was not the idea that created the modern cylindrical grinding machine, although it was the idea that created the original cylindrical grinding machine. The modern grinding machine was originated to assist the lathe in creating cylindrical work and to make the process less expensive than it had been before any grinding was done—not simply to replace filing.

The suggestion of Mr. Kent is a good one. In Par. 35 are given all the details in connection with grinding certain shafts, though not in tabular form. These are what might be called countershafts. The cost of the wheels is given, the wear of the wheels, the cost of power, time and amount of steel removed with every cubic inch of wheel, etc.

No. 1370

INCREASE OF BORE OF HIGH-SPEED WHEELS BY CENTRIFUGAL STRESSES

WITH FORMULAE FOR FORCE AND SHRINK FITS

By Sanford A. Moss, West Lynn, Mass.

Member of the Society

Recent advances in high-speed steam turbines and the apparatus driven by them have opened up a great many new problems. One of these is the fact that, under certain circumstances, unless proper precautions are taken, a high-speed turbine wheel, or the like, expands at the hub a sufficient amount to cause an appreciable increase in the bore, so as to make the wheel slightly loose upon the shaft and capable of pounding back and forth. This pounding causes a mechanical increase of the looseness, so that the machine soon gives evidence of distress.

2 So far as the writer knows, the first one to direct attention to this problem, to suggest means of investigating it, and to offer a solution, was Richard H. Rice, chief engineer of the Lynn works' turbine department, General Electric Company. The present paper gives an account of work on the problem done under the general direction of Mr. Rice by various members of his department.

COMPUTATION OF INCREASE OF BORE OF A ROTATING WHEEL

NATURE OF STRESSES IN A WHEEL OR RING

3 It is the purpose first to give a general account of methods of computing stresses in circular bodies. When there is rotation this becomes a very intricate subject and details and formulae must be sought in works specially devoted to it.

4 It is assumed that the disk is thin, that is, that the thickness is negligible compared with its diameter. Then conditions are the same at all points on a line parallel to the axis. This is not

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strictly true, as the central part of a hub is strained differently from the ends. However, in most cases this effect is slight.

5 At each radius in the disk there are two stresses, one in the radial direction denoted by S_r and one in the tangential direction denoted by S_t. If we consider the effect of these stresses on the metal of a thin ring at the interior of the disk, we find the circumference of the ring to be increased mainly by virtue of the tangential stress, and the thickness mainly by the radial stress. However, when a metal is extended in one direction by a stress there is a shrinkage in a direction at right angles, the ratio of the deformations being called "Poisson's ratio," which we will denote by V. For ordinary kinds of steel this has a value of 0.3. That is, the increase of circumference of a thin ring in the interior of the disk is less than that which would be produced by the tangential stress if acting alone, by an amount which is 0.3 of the extension which would be produced by the radial stress acting alone. Similarly the increase in the thickness of the ring is less than that produced by the radial stress acting alone. In any case, the ratio of stress to extension per unit length produced by it if acting alone, is given by the modulus of elasticity E, usually taken as 29,000,000 lb. per sq. in.; or, if l is the total extension in a distance L, we have the well-known law

$$E = \frac{S}{\frac{l}{L}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

The combination of the above principles with the laws of the centrifugal forces on the various elements of the disk gives formulae which enable computations to be made of the stresses in the disk at all points during rotation.

6 In order to determine the safety of a structure where there are stresses in more than one direction, it is necessary to find an effective stress which corresponds so far as yielding is concerned to the stresses in a specimen of the same material subject to simple tension in a testing machine. It is now almost universally agreed that Guest's maximum shear law is the proper one. This states that in cases such as the present, where there are stresses in two directions only with zero stress in the third, and where the stresses are either both tension or both compression, the greater of the two stresses is the effective stress. This effective stress is equal to a simple stress which would produce like results so far as yielding, safety, elastic limit, etc., are concerned.

7 This is not true, however, when there are stresses in all three di-

rections, as in the case with a rotating disk of appreciable thickness where there is change of stress in the axial direction; or with two stresses with opposite sign, as in the stresses in the hub due to force fits, discussed in Pars. 29-39. In such cases the maximum algebraic difference between the greatest and least of the stresses existing at a point taken with proper sign gives the effective stress or equivalent simple tension.

8 In most cases of properly designed rotating wheels with a hole at the center the maximum stress is the tangential stress at the bore. This is, therefore, the stress which is to be considered in order to ascertain if the wheel is safe and is further the stress which must be determined in order to discuss the problem of the present paper.

FORMULA FOR INCREASE OF BORE

9 As we proceed outward from the bore in a rotating wheel, the radial stress mounts up rapidly, due to the fact that the centrifugal forces of the outer portions of the disk pull outward on the inner portions. The nearer the center the less the material in the inner circle, and therefore the less resistance to radial elongation. At the bore there is no force whatever resisting radial elongation, so that the radial stress becomes zero as this inner diameter is reached. We have then on the inner wall of the cylinder forming the bore, material which is stressed only in the tangential direction and the increase of circumference is exactly the same as if we had a straight bar of the same length stressed an amount equal to the tangential stress at the bore. This is expressed by the simple relation which is the fundamental equation of our problem:

$$X = \frac{S_{to}\, D_o}{E} \qquad [2]$$

where

X = increase in diameter in inches of the bore, due to the centrifugal stresses at a given speed

S_{to} = tangential stresses at the bore in pounds per square inch at the same speed

D_o = original diameter of bore in inches

E = modulus of elasticity, 29,000,000 for steel

10 It can be shown that the stresses at any point in a given wheel or at a similarly situated point in any other wheel of similar proportions, are directly proportional to the square of the peripheral wheel speed. Hence the increase of bore in which we are interested varies directly with the square of the peripheral wheel speed with

a given wheel. In any case of the same or geometrically similar wheels, the increase per foot of bore increases directly with the square of the wheel speed.

NUMERICAL MAGNITUDE OF INCREASE OF BORE

11 The allowable values of the tangential stress at the hub are quite high, since a centrifugal load is one of the easiest loads which can be applied and is of the nature of a pure dead load in a beam. It is not possible to apply or remove a centrifugal load suddenly,

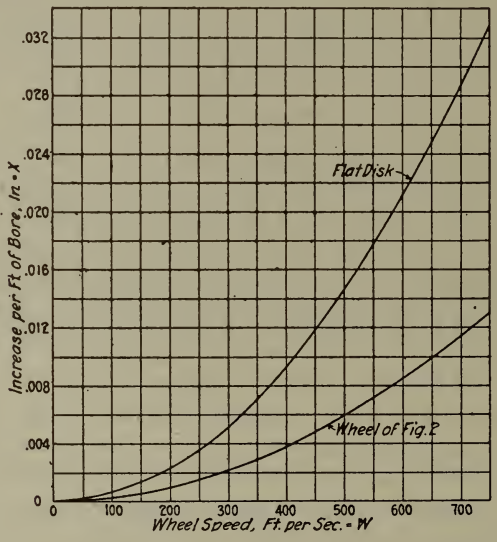


FIG. 1 INCREASE OF BORE OF A ROTATING WHEEL

Ratio of Diameter of Bore to Diameter of Disk = 0.353. This Ratio may be changed considerably without appreciably affecting the Results.

Approximate Dead Weight per Pound of Disk (equal in Both Cases) = 100,000 Lb. per Ft. of Disk Circumference at 534 Ft. per Sec.

but only in the most gentle manner. Also the maximum stress usually occurs only in a single set of fibers which are reinforced by adjacent fibers with decreasing stresses. There need be no "factor of ignorance" and maximum stresses can be known with as great accuracy as it is desired to give to the mathematical computations. In many cases 20,000 lb. per sq. in. is a conservative stress. In some cases with special grades of steel 30,000 lb. per sq. in. is possible. For a wheel with 12-in. bore and 30,000-lb. maximum stress the increase in bore computed from [2] is 0.0124 in., which is an appreciable

amount. For other stresses and other dimensions of bore the increase of bore is in direct proportion. When the stress is 29,000 lb. per sq. in. the increase in bore is 1 mil per in. of bore.

12 Fig. 1 gives numerical values of increase in bore for various wheel speeds for two different shapes of disk. The upper curve is for a disk of uniform thickness. This shape is only possible with comparatively low wheel speeds and the stresses become excessive with speeds ordinarily used in turbine practice. The lower curve is for a disk of the more efficient shape shown in Fig. 2. In both cases the computations are made for a turbine bucket wheel with a load

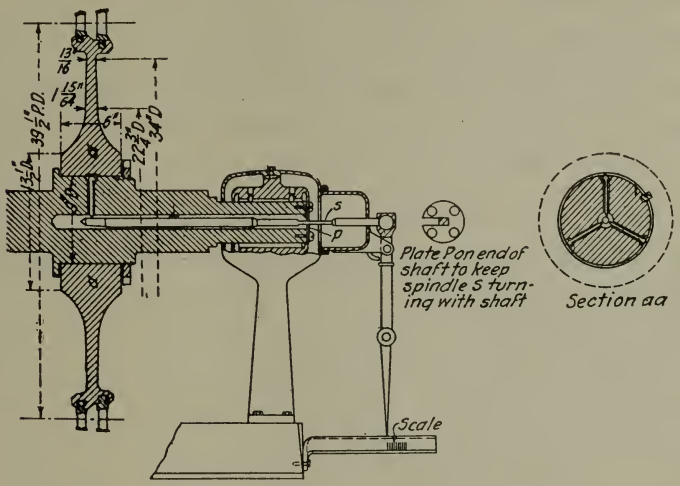


FIG. 2 DIAGRAM OF DEVICE FOR MEASURING THE INCREASE OF BORE DURING ROTATION

due to centrifugal force of buckets at 3600 r.p.m. of 100,000 lb. per ft. of circumference. The shapes of the curves are affected somewhat by the magnitude of this outer dead load. In both cases the ratio of bore to disk diameter is 0.353. The curves are not greatly affected by difference in this ratio. The curves give very nearly the values of increase in bore per foot of bore for various wheel speeds for a disk of uniform thickness, which is about as poor a shape as would ever be used, and for a disk of what is probably as efficient a shape as would ever be used, with an average bucket load. The increase in bore for any shaped disk whatever will probably lie between the two curves.

13 The curves of Fig. 1 were constructed as follows: The wheel

whose dimensions are given in Fig. 2 is the wheel of a standard steam turbine with a web diameter of 34 in. In the course of the design of this wheel the maximum stress was computed for the rated speed, 3600 r.p.m., and came out about 20,400 lb. per sq. in. From the value of this stress and formula [2], the increase in bore at the rated speed was found. The effective bore, taking account of keyways, was 12 in. From the law that the stress and hence the increase in bore varies directly with the square of the wheel speed, the lower curve of Fig. 1 was drawn, taking bore as 12 in., however.

14 Next was taken the case of the upper curve for a flat disk. From the formulae for stresses in flat disks given in Stodola's "Die Dampfturbinen," fourth edition, pp. 248 and 249, can be deduced the following formula for the tangential stress at the bore of a flat disk S_{to}, where S_{rl} is the radial stress at the circumference due to bucket load, W is the wheel speed in feet per second and R is the ratio of bore diameter to web diameter

$$S_{to}=0.0261W^2\ (0.7R^2+3.3)+\frac{2S_{rl}}{1-R^2}\ldots\ldots\ldots\ldots[3]$$

15 This is for a steel disk whose modulus of elasticity is 29,000,000 lb. per sq. in., Poisson's ratio 0.3 and density 0.28 lb. per cu. in.

16 The centrifugal force due to the bucket load on the previous disk, 100,000 lb. per ft. of circumference, was taken as being applied to a flat disk of about the same weight, giving $1\frac{1}{4}$ in. axial width, and 34 in. web diameter. This gives a radial stress S_{rl} of 6670 lb. per sq. in. at the circumference of the flat disk. The ratio of the diameter of bore to the web diameter R was taken as in the previous disk 0.353.

EXPERIMENTAL MEASUREMENT OF INCREASE OF BORE OF A ROTATING WHEEL

17 The magnitudes of the increase of bore in a number of actual turbine wheels, as determined by substituting computed stresses in equation [2] came out so large that it was decided to make an actual measurement of the increase in bore when a wheel was rotating.

18 Fig. 2 shows the apparatus diagrammatically and Fig. 3 the results obtained. The turbine wheel was mounted on a shaft with a light force fit. A delicate micrometer arrangement was provided so that change in bore of the wheel while it was rotating with the shaft could be observed. The wheel was gradually increased in speed and meanwhile the readings of the micrometer taken. These

showed that the bore expanded at each speed an amount almost exactly the value computed by the methods previously given. The micrometer arrangement proved very positive and there was no uncertainty regarding the measurement of the increase of bore. The maximum amount measured was about 0.007 in. The observations were accurate to about 0.0005 in.

19 The wheel was forced on the shaft with a force fit of 0.0014 so that it was loose on the shaft by about 0.0056 at maximum speed. The duration of the tests was not great enough to show any difficulty due to this.

MEASURING DEVICE

20 The measuring device shown in Fig. 2 consisted of three pistons sliding in radial holes in the shaft pushed outward by means of a tapered plug moving axially in a hole in the center of the shaft. The motion of the central plug was given by means of a lever with a ball and socket joint fulcrumed so as to give considerable multiplication and read by a pointer passing over a scale. A displacement of the pointer by 1 in. corresponded to an increase of bore of 0.004 in. The position of the pointer was certain to within about $\frac{1}{8}$ in. motion along its scale, corresponding to 0.0005 in. change of bore. While the speed was being varied an attendant kept his hand on the pointer, moving it back and forth a slight amount so as always to keep it just in the proper position. As the speed varied from low values to maximum value the pointer moved a distance of about 2 in. so that the change in bore was quite perceptible.

21 Owing to the fact that the increase of bore varies with the square of the speed there was no perceptible change below about 750 r.p.m., and this was the lowest speed used. The highest speed was 3900 r.p.m.

CALIBRATING SLEEVE TEST

22 In order to find the relation between the readings of the pointer and the actual values of increase of bore, and also to demonstrate that the apparatus would give correct measurements at high speed, a calibrating sleeve was made. This was a thin sleeve bored out at each end so as to be a sliding fit on the shaft and with two different bore diameters in the central portion, either of which could be brought over the pistons of the measuring device. Readings of the pointer were taken at each of these bores, stationary and rotating, at various speeds. The stationary readings gave the motion of the pointer for the known change of bore calibrating the

entire apparatus. The readings while rotating showed that the instrument gave correct readings at all speeds.

TEST WITH ROTATING WHEEL

23 Next the rotating wheel was pressed on the shaft. A number of readings of the diameter of shaft and bore of wheel showed that the average amount of force fit was 0.0017 in. After this, a pair of outside calipers was set to fit the shaft at various points and a pair of inside ones set to fit the wheel bore. The difference

TABLE 1 READINGS OF POINTER IN TESTS WITH ROTATING WHEEL, CONDUCTED MARCH 11, 1911

	Test No. 9		Test No. 10		Test No. 11		Test No. 12		Test No. 13	
R.P.M.	Up	Down	Up	Down	Up	Down	Up	Down	Up	Down
0	2.61	2.62	2.62	2.63	2.62	2.63	2.63	2.62	2.61	2.62
750	2.61	2.54	2.63	2.59	2.63	2.59	2.64	2.62	....	....
1050	2.64	2.58	2.70	2.68	2.68	2.67	2.66	2.70	....	....
1200	2.69	2.63	2.74	2.72	2.72	2.74	2.72	2.75	....	....
1500	2.72	2.70	2.80	2.82	2.82	2.80	2.84	2.90	....	....
1800	2.78	2.79	2.90	2.96	2.92	2.90	2.96	3.00	....	....
2100	2.84	2.89	3.03	3.10	3.02	3.10	3.11	3.20	....	....
2400	2.99	3.14	3.18	3.25	3.18	3.24	3.26	3.40	....	....
2700	3.16	3.30	3.32	3.44	3.32	3.48	3.48	3.56	....	....
3000	3.30	3.45	3.54	3.63	3.54	3.63	3.70	3.75	....	....
3150	3.50	3.64	3.70	3.78	3.75	3.85	3.85	3.85	....	....
3300	3.65	3.70	3.85	3.94	3.85	3.98	4.00	4.00	....	....
3450	3.80	3.80	4.01	4.08	4.00	4.08	4.10	4.08	....	....
3600	4.00	4.00	4.13	4.20	4.15	4.18	4.24	4.20	....	....
3750	4.12	4.12	4.24	4.28	4.30	4.29	4.34	4.33	....	....
3900	4.30	4.30	4.34	4.34	4.38	4.38	4.38	4.38	4.38	4.38

between the two calipers was then found by means of a thickness gage. The mean of these observations gives 0.0011 in. as the amount of force fit. The mean of these two values 0.0014 in. is considered to be the actual amount of force fit. It was necessary to know the amount of this force fit in order to make comparisons between the computed and observed amounts of increase of bore of the rotating wheel, since the wheel at zero speed is expanded somewhat.

24 Fig. 2 is a diagram of the arrangement of the apparatus when the wheel was on the shaft. Readings of the pointer were taken at various speeds as shown by Table 1 and the results plotted in Fig. 3. The pointer readings, as well as the corresponding values of actual increase in bore as shown by the previous calibrations, are given

on the left. Owing to the fact that the wheel was forced on the shaft, the measured diameter of the wheel at zero speed is greater than the actual free diameter of the wheel. By methods given in Pars. 29-39, it was computed that the wheel was expanded by the force fit one-half of the difference in free diameters, the remainder being compression of the shaft. Hence the base line for increase in bore of the wheel from the free condition is the lower line of Fig. 3, which is 0.0007 in. below the base line for increase in bore from the

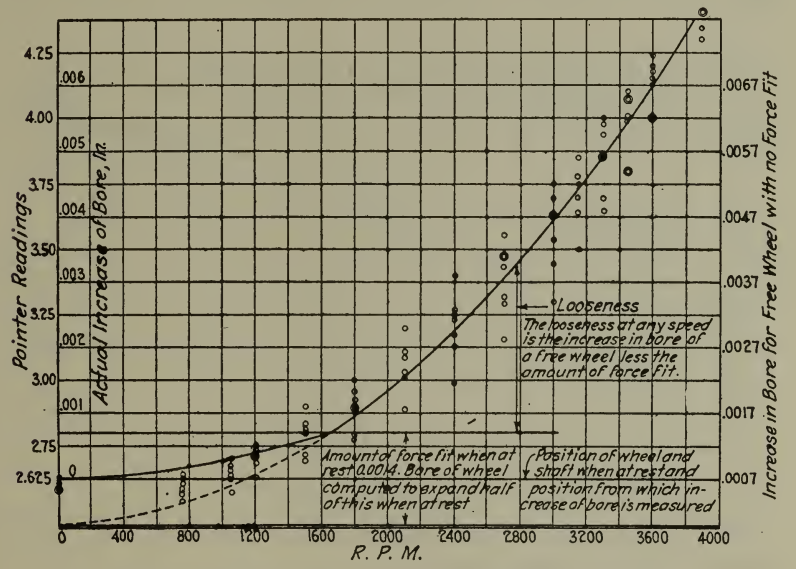


FIG. 3 INCREASE IN BORE OF A ROTATING WHEEL
Points from Test with Rotating Wheel (Table 1); Curve drawn from Computations.

actual initial value when on the shaft. The computed curve for the increase in bore for the free wheel at the various speeds is drawn in Fig. 3 beginning with the lower base line. At that speed where the expansion of the free wheel is equal to the amount of the force fit, 0.0014 in., the shaft is just released from its compression and wheel and shaft are just of the same diameter. For speeds beyond this point the wheel expands exactly as if there was no initial force fit. The wheel is loose on the shaft by whatever increase in bore occurs beyond this point. The observed points are seen to fit very closely the computed curve. For all speeds below that at which the wheel and shaft just touch there is compression of the shaft and expansion

of the wheel by the force fit, the amount being successively less as the speed increases. The effective force fit at any speed is the difference between the force fit at rest, and the expansion of the free wheel at the given speed. The effective force fit is, therefore, the difference between the dotted curve and the upper horizontal line. As shown later, when there are different amounts of force fit with a given wheel and shaft, the expansion of the wheel is always a constant fraction of the amount of force fit. This is one-half in the present case. Hence the curve showing the increase in bore of the wheel,

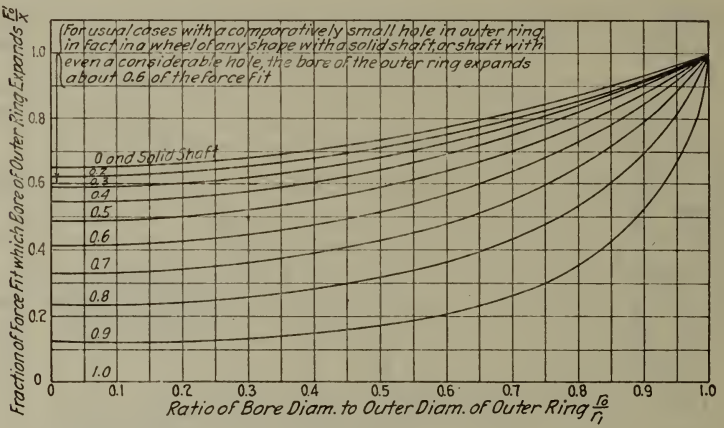


FIG. 4 RATIO OF EXPANSION OF BORE TO FORCE FIT

Figures on Curves denote Ratio of Diameter of Hole in Inner Ring to Diameter of Bore of Outer Ring $\frac{r_2}{r_0}$

as the force fit is gradually relieved, is drawn halfway between the dotted curve and the upper horizontal line.

25 In the present case the amount of the force fit is comparatively small. It was originally intended to repeat the experiments with successively increasing amounts of force fit. The measuring apparatus was arranged so that it could be torn down and reassembled without change in the absolute initial value of the readings. This was done with the idea actually of measuring the change in the initial bore of the wheel when put on the shaft with different amounts of force fit, giving curves in Fig. 3 which would start with successively greater amounts of force fit and meet the curve of a free wheel higher up. However, the results of the first experiment agreed so closely with the theory and everything was

so obvious after study of the subject that no further experiments were made.

DESIGN OF WHEELS SUBJECT TO INCREASE OF BORE BY CENTRIFUGAL STRESSES

26 The method used to prevent difficulty due to loose wheels is to force the wheel on the shaft with a force fit, or difference in diameters between wheel bore and shaft, greater than the computed amount of increase of bore at maximum speed. In some cases this has resulted in large amounts of force fit requiring very heavy pressures for forcing. Special material is necessary for the bushings between the wheel and the shaft. However, all of these points have been successfully met and ill effects due to loosening of the wheels wholly eliminated.

27 In some cases the wheel hubs do not fit along their entire length but only in places. The radial pressure when the fit only occurs for a short distance is obviously increased by the ratio of the total length of hub to the actual length of the fit. The numerical values of this radial pressure on the bushings, computed by methods discussed in Par. 33, come to very high figures in many cases. Of course, some relief is offered by the fact that this pressure is pure compression and that the metal of the bushings is constrained on nearly all sides so that the resistance to flow is considerable. Under such circumstances much higher values of compression stress can be used than when there is no resistance to the deformation which the compression tends to cause. There is reason to believe that in the ideal case of a piece subject to pure compression in every direction, there would be no failure even with an infinitely large stress.

28 It is to be remarked that when the force fit is just sufficient to prevent loosening at full speed and the shaft is solid, the effective stress in the hub due to the force fit when at rest, is exactly equal to the stress at full speed.

COMPUTATION OF FORCE AND SHRINK FITS WITH STEEL

29 A complete treatment of the subject based on the maximum shear law previously mentioned is given for the first time so far as known. The general nature of ring stresses is in Pars. 3-8. The mathematical derivation of the formulae is given in the Appendix.

30 It turns out that a wheel hub is affected only for a very short distance outward so that the nature of the outer portion has little or no influence. The condition near the hub in a flat disk with an

outer diameter only ten times the diameter of the bore is practically the same as in a disk with an infinitely large outer diameter. We therefore consider only the case of a ring forced upon another ring of equal axial length. We estimate the outer diameter of the outer ring so as to make it equivalent so far as expansion is concerned to a given wheel of any shape. For instance, for a solid wheel comparatively thin at the outer portion, we would estimate the equivalent flat disk to have one-half or one-third the diameter of the wheel.

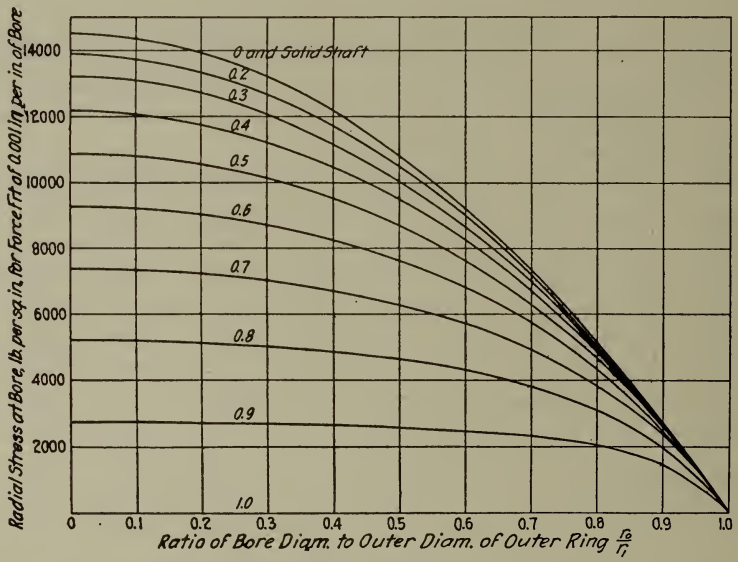


FIG. 5 RADIAL STRESS AT BORE S_{r0}. FORCE FIT OF 0.001 IN. PER IN. OF BORE

Stress is directly proportional to Force Fit per Inch of Bore. Figures on Curves denote Ratio of Diameter of Hole in Inner Ring to Diameter of Bore of Outer Ring $\frac{r_2}{r_0}$.

For a wheel with spokes, we have little more than the hub only. We also estimate an equivalent inner ring if we do not have a solid or hollow shaft. Owing to the comparatively slight influence of the parts of the rings furthest from the fit, a considerable error in estimation of the diameters of the equivalents will cause comparatively small error in the results.

31 Figs. 4, 5 and 6 give amount of expansion and stresses in an outer ring or hub pressed or shrunk on inner ring or shaft.

32 The difference in the free diameters is called the force fit, which we will denote by X. When the hub is in place on the shaft and there is no rotation so as to loosen, it is expanded by a certain amount F'_o. The shaft is also compressed by a considerable amount. The sum of the increase of hub bore, F'_o, and the compression of the shaft is evidently the amount of the force fit. Fig. 4 gives values of $\frac{F'_o}{X}$ the fraction of the force fit which the hub expands.

33 Fig. 5 gives the radial stress at the bore of the hub, which is

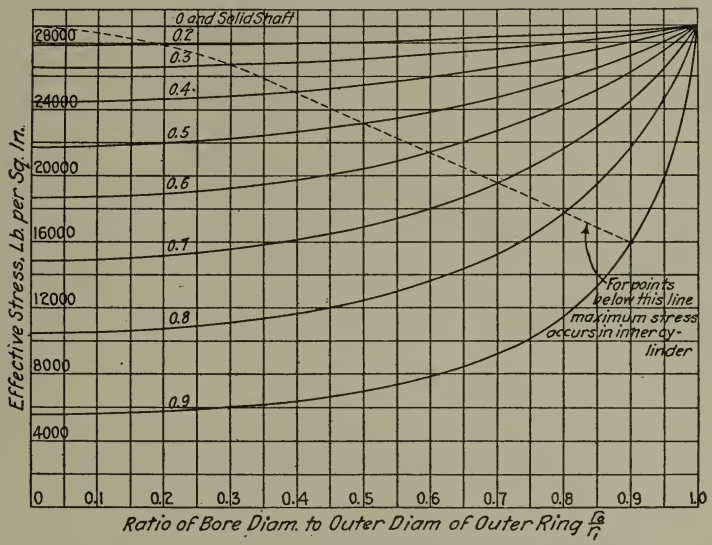


FIG. 6 EFFECTIVE TENSILE STRESS AT BORE OF OUTER RING. FORCE FIT OF 0.001 IN. PER IN. OF BORE

Stress is directly proportional to Force Fit per Inch of Bore. Figures on Curve denote Ratio of Inner Diameter of Inner Ring to Bore Diameter $\frac{r_2}{r_0}$.

also the stress at the outside of the shaft, as well as the pressure per square inch between when there is actual contact for the full length of hub.

34 Fig. 6 gives the maximum effective stress at the bore of the hub. This is the greatest stress in the system unless the shaft is hollow and relatively thin. Figs. 5 and 6 are for a force fit of one mil per inch of bore. Stresses for other values are in proportion.

35 The abscissae of these figures give the ratio of the bore to the outer diameter of the hub. The upper curve in each of these

three figures is for a solid shaft or one with a small hole and the other curves are for rings with various sizes of holes. When the abscissa is zero the outer diameter of the hub is infinitely large as compared with the bore. The curves are nearly horizontal for a considerable distance in this region and the results for an outer diameter ten times the diameter of the bore, that is, a ratio of 0.1, are almost the same as for an infinite outer diameter. The values

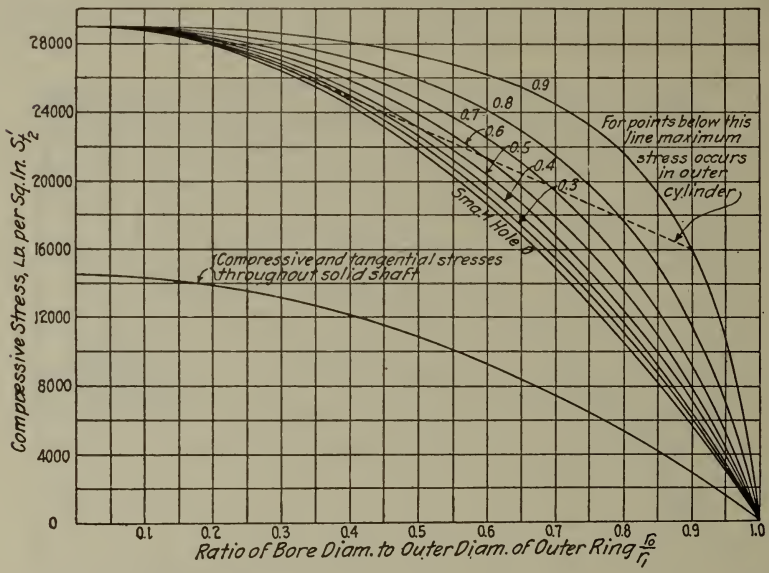


FIG. 7 COMPRESSIVE TANGENTIAL STRESS S'_{t2} AT INNER DIAMETER OF INNER RING. FORCE FIT OF 0.001 IN. PER IN. OF BORE

Stress is directly proportional to Force Fit per Inch of Bore. Figures on Curves denote Ratio of Inner Diameter of Inner Ring to Bore Diameter $\frac{r_2}{r_0}$. Values given are the Maximum Effective Stresses in Inner Ring.

for an outer diameter five or three times the diameter of the bore change but little.

36 Similarly, as will be seen by noting the curves for the various values of the ratio of the diameter of the hole in the shaft to the shaft diameter, the hole has but little influence unless it is comparatively large. The usual case of a shaft with a comparatively small hole is practically equivalent to a solid shaft. As will be seen by a study of the curves, the case where the diameter of the hub is seven times the diameter of the bore, giving a ratio of 0.14, and

where the hole in the shaft is 0.3 of the shaft diameter, gives very nearly the results of any usual case. Hence, for usual cases the hub bore expands about 0.6 of the amount of the force fit and for a force fit of one mil per inch of bore the radial stress is about 13,000 lb. per sq. in. and the effective tangential stress is about 27,000 lb. per sq. in.

37 For a solid shaft, however, the maximum effective stress is 29,000 lb. per sq. in. for a force fit of 1 mil per in. of bore, regardless of the character of the wheel or hub. The stress for other amounts of force fit is in proportion.

38 Force fits of 1 mil per in. are therefore about large enough to give no permanent set of the hub for steel and a solid shaft. Force fits up to about $1\frac{1}{2}$ mils are frequently used. These give some set of the fibers very near the bore. This set is almost entirely eliminated by the elastic force of the outer fibers if the hub is ever removed from the shaft, however.

39 Fig. 7 gives the maximum stress (at the inside) in the shaft or inner ring. This is usually less, but for a relatively thin ring may be greater, than the stress in the hub. A solid shaft has a stress one-half as great as one with a small hole.

40 A rational expression for the force in pounds to press the hub on the shaft is

$$0.12\, d_o l S_{ro}$$

where l is the length, d_o the diameter of bore of the hub in inches and S_{ro} the radial stress at the bore found from Fig. 5. The coefficient of 0.12 was obtained from actual tests and gives a coefficient of friction of 0.038.

41 For a force fit of one mil per inch of bore the forcing pressure in our usual case is about 1560 lb. per in. of bore and per in. of hub length.

42 The forcing pressure of course varies widely on account of the influence of surfaces and lubricant. The formula gives an average value.

APPENDIX

DERIVATION OF FORMULAE FOR FORCE AND SHRINK FITS [1]

NOTATION

S_r = Radial stress, lb. per sq. in.
S_t = Tangential stress, lb. per sq. in.
r = Radius, inches
E = Modulus of elasticity taken 29,000,000 lb. per sq. in.
V = Poisson's ratio, taken 0.3
X = Amount of force fit, or difference in diameters, inches
F = Increase of diameter, inches

40 We take the case of two steel rings, the outer one being forced on the inner one. A subscript after any symbol denotes the value for a particular radius: 1 denotes the outer diameter of the outer ring or hub; 0 denotes the common diameter where the force fit occurs and 2 denotes the bore of the inner ring. We will also use an index $'$ to denote values for the outer ring in general and indexes $''$ to denote the inner ring.

41 The increase in radius of an elementary ring is $\frac{F}{2}$ and the tangential strain is

$$\frac{F}{2r}=\frac{S_t-VS_r}{E}$$

The radial strain is

$$\frac{1}{2}\frac{dF}{dr}=\frac{S_r-VS_t}{E}$$

and by using the preceding value of F we obtain a fundamental equation applying to any circular body

$$V\left(S_r+r\frac{dS_r}{dr}-S_t\right)=S_t+r\frac{dS_t}{dr}-S_r \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

42 The statement that the radial forces on an elementary segment in a stationary cylinder are balanced is

$$S_r dr\, d\theta + r\, d\theta\, dS_r - 2\, S_t dr \sin d\frac{\theta}{2}=0$$

This reduces to a second fundamental differential equation

$$S_r+r\frac{dS_r}{dr}-S_t=0 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [5]$$

This enables reduction of the first fundamental equation to the form

[1] This section was written with the assistance of Mr. C. A. Schellens of the General Electric Company, Lynn, Mass.

$$S_t+r\frac{dS_t}{dr}-S_r=0 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [6]$$

We integrate these simultaneous differential equations by adding and integrating, giving

$$S_r+S_t=2A$$

We add this to the second fundamental equation which separates the variables. By multiplying by r we obtain the exact differential of r^2S_r so that we obtain the integral directly. From the equation above we obtain the other integral, giving the equations

$$S_r'=A'-\frac{B'}{r^2}$$

$$S_t'=A'+\frac{B'}{r^2}$$

$$S_r''=A''-\frac{B''}{r^2}$$

$$S_t''=A''+\frac{B''}{r^2}$$

43 A' B' A'' B'' are four constants to be determined by the boundary conditions. Two of these are that the radial stresses at the inner and outer radii r_1 and r_2 are zero for the case of a hole in the inner ring. That is

$$S'_{r1}=0$$

$$S''_{r2}=0$$

Substituting these values in the above equations

$$0=A'-\frac{B'}{r_1^2}$$

$$0=A''-\frac{B''}{r_2^2}$$

44 This enables us to eliminate two constants giving a new set of fundamental equations

$$S_r'=B'\left(\frac{1}{r_1^2}-\frac{1}{r^2}\right)$$

$$S_t'=B'\left(\frac{1}{r_1^2}+\frac{1}{r^2}\right)$$

$$S_r''=B''\left(\frac{1}{r_2^2}-\frac{1}{r^2}\right)$$

$$S_t''=B''\left(\frac{1}{r_2^2}+\frac{1}{r^2}\right)$$

45 The third boundary condition is that the radial stresses at the common surfaces are equal, that is

$$S'_{r0}=S''_{r0}$$

giving

$$B'\left(\frac{1}{r_1^2}-\frac{1}{r_0^2}\right)=B''\left(\frac{1}{r_2^2}-\frac{1}{r_0^2}\right)$$

46 The fourth condition is that the algebraic difference between the deformations at the common radius is equal to the force fit. That is

$$F_0'-F_0''=X$$

47 In general the radial deformation at any radius is given by

$$\frac{F}{2}=\frac{r}{E}\left(S_t-VS_r\right)$$

Outward deformation and tensile stress are counted positive.

48 Then we have, after noting that $S'_{r0}=S''_{r0}$

$$\frac{2r_0}{E}\left\{S'_{t0}-S''_{t0}\right\}=X$$

Substituting the stresses from the last set of equations and substituting for B'' from the equation given by the third boundary condition, this becomes

$$B'\left\{\frac{1}{r_1^2}+\frac{1}{r_0^2}-\frac{\left(\frac{1}{r_2^2}+\frac{1}{r_0^2}\right)\left(\frac{1}{r_1^2}-\frac{1}{r_0^2}\right)}{\left(\frac{1}{r_2^2}-\frac{1}{r_0^2}\right)}\right\}=\frac{EX}{2r_0}$$

Hence

$$B'=\frac{EX}{4r_0}\,\frac{r_1\left(1-\frac{r_2^2}{r_0^2}\right)}{\frac{r_1^2}{r_0^2}-\frac{r_2^2}{r_0^2}} \qquad [7]$$

This gives a value of B' which we can substitute in the fundamental equations. The value of the radial stress at the common radius then comes out

$$S'_{r0}=-E\frac{X}{4r_0}\,\frac{\left(\frac{r_1^2}{r_0^2}-1\right)\left(1-\frac{r_2^2}{r_0^2}\right)}{\frac{r_1^2}{r_0^2}-\frac{r_2^2}{r_0^2}} \qquad [8]$$

This is a compressive stress. The stress varies in the outer ring from the above value at the bore to zero at the outside.

49 Fig. 5 gives values computed from this equation. Values of radial stress S_{r0} are ordinates, for a value of $\frac{X}{2r_0}$, the force fit per inch of bore, of one mil per inch. Values of $\frac{r_0}{r_1}$ are abscissae. Each of the curves is for a given value of $\frac{r_2}{r_0}$.

50 The fraction of the force fit which the outer ring expands is given by

$$\begin{aligned}\frac{F_0'}{X}&=\frac{2r_0}{EX}\left(S'_{t0}-VS'_{r0}\right)\\&=\frac{2r_0B'}{EX}\left\{\frac{1}{r_1^2}+\frac{1}{r_0^2}+V\left(\frac{1}{r_0^2}-\frac{1}{r_1^2}\right)\right\}\\&=\frac{1}{2}\,\frac{\left\{1+\frac{r_1^2}{r_0^2}+V\left(\frac{r_1^2}{r_0^2}-1\right)\right\}\left(1-\frac{r_2^2}{r_0^2}\right)}{\frac{r_1^2}{r_0^2}-\frac{r_2^2}{r_0^2}}\end{aligned} \qquad [9]$$

51 Fig. 4 gives values computed from this equation. The ordinates are values of $\frac{F'_0}{X}$, the fraction of force fit which the outer ring expands. Values of $\frac{r_0}{r_1}$

are abscissae. Each of the curves is for a given value of $\frac{r_2}{r_0}$.

52 If we substitute the value of B' in the last set of fundamental equations and obtain the value of S'_t we find that there is a tangential tensile stress in the outer cylinder with greatest value at the bore. Hence, since the radial stress at this point is compressive, we have as the greatest value of the effective stress in the outer cylinder, at the bore

$$S'_{t0} - S'_{r0} = E\frac{X}{2r_0}\frac{1-\frac{r_2^2}{r_0^2}}{1-\frac{r_2^2 r_0^2}{r_0^2 r_1^2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [10]$$

53 Fig. 6 gives values computed from this equation. For a small hole in the inner cylinder, and also, as will be shown later, for a solid shaft, this maximum stress in the outer cylinder, at its bore, becomes

$$E\frac{X}{2r_0}$$

This value of the maximum effective stress in a hub on a solid shaft, obtained in the correct way by adding the tangential stress due to hub extension and radial stress due to shaft compression, turns out to give the same final results as the incorrect method of finding the tangential stress in the hub with an incompressible shaft.

54 From the relation between B' and B'' and the value of B' we can substitute in the last set of fundamental equations and obtain the stresses in the inner ring. Both stresses are compressive throughout so that the greatest gives the maximum effective stress direct.

55 The radial compressive stress increases from zero at the inner diameter to a maximum at the outer diameter of the inner ring, where it has the value already given for the bore of the outer cylinder and shown in Fig. 5. The tangential compression stress increases from the outer diameter to the diameter of the central hole where it is the maximum stress in the inner cylinder and is

$$S''_{t2} = -E\frac{X}{2r_0}\frac{1-\frac{r_0^2}{r_1^2}}{1-\frac{r_2^2 r_0^2}{r_0^2 r_1^2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [11]$$

Values of the expression are plotted in Fig. 7. It is to be noted that in case the ratios of radii of both rings are alike, the above expression for maximum stress in the inner ring coincides with the previous expression for maximum effective stress in the outer ring. It can also be seen that the greater of the two maximum stresses is in that ring which has the greatest ratio of inner to outer diameter, i. e., which is relatively thinnest.

56 The preceding work has been for the case where there was a hole in the inner cylinder. For a solid shaft, the inner boundary condition in the inner cylinder becomes

$$S''_{t2} = S''_{r2}$$

instead of

$$S''_{r2} = 0$$

the other conditions remaining the same. From the first set of fundamental equations and the above relation

$$A''r_2 - B'' = A''r_2 + B''$$

If we now put $r_2 = 0$ we have

$$B'' = 0 \text{ and } S''_r = S''_t = A''$$

That is, the radial and tangential stresses are equal and constant throughout a solid shaft. This is a well-known property of any solid with uniform normal stress over the entire surface.

57 By inserting the remaining boundary condition we obtain final values for the stresses in the outer cylinder exactly the same as the previous values for a hollow shaft when the radius of the inner hole r_2 is made equal to zero, giving an infinitely small hole. The value of the uniform stress throughout the shaft is

$$S''_r = S''_t = -\frac{E}{2}\frac{X}{2r_0}\left(1 - \frac{r_0^2}{r_1^2}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [12]$$

58 Values of this expression are given by the lower curve of Fig. 7. It is exactly one-half the previous value of the stress at the center of a ring when the radius of the inner hole r_2 is made equal to zero, giving an infinitely small hole.

DISCUSSION

Richard H. Rice said that the paper showed, first, that the computations made for stresses in wheels running at high speed are correct. The forms of wheels used in turbines, in air compressors and other high-speed machinery are complex, as are also the analyses for the solution of the stresses and strains in such wheels. It is gratifying to find that actual measurements on the many wheels measured confirm accurately our computations and show that the formulae are adequate for the purpose.

The paper also shows that wheels running at high speeds do expand in the bore, although some little question has been raised as to this, and the amount of expansion can be accurately computed. It also shows that such expansion can be adequately taken care of and that it is a matter of no importance, so long as the designer knows the amount of expansion so that he can provide a forced fit somewhat greater than the expansion.

Wm. H. Kenerson said that it would help in an interpretation of the results of the tests described in the paper to know something of the length and character of the end of the little piston pieces running from the sleeve in the shaft to the hole whose diameter is being measured, and he hoped the author would tell something about this, to furnish an idea of how much that distance changes when the shaft is going at high speed.

Mr. Rice replied to Professor Kenerson's question that that point is provided for in the preliminary calibrations where the readings were calibrated by working on sleeves of known diameter. In reply to a further question he added that the sleeves provided were of such small diameter their expansion was negligible.

Wm. Kent asked how Poisson's ratio had been determined, and said that while he had seen many figures published of Poisson's ratio, he had never seen a statement of how it was measured. He desired to know whether there were any American experiments on the subject to determine Poisson's ratio.

Elmer A. Sperry wrote that he was personally interested in the paper, as he had himself pursued similar investigations especially upon the subjects treated in the first part of the paper, and apparently had carried them even further than the author.

As a basis of two large gyroscopes which together constitute

TABLE 2 VALUES OF POISSON'S RATIO (RATIO OF CROSS CONTRACTION TO DIRECT EXTENSION PRODUCED BY A SIMPLE TENSION)

Experimenter	Reference	Material	Young's Modulus	Poisson's Ratio	Method	Measurement
Wertheim	Annales de Chimie et de Physique, ser. 3, vol. 23, 1848.	brass		0.333	Tube closed at both ends subjected to longitudinal tension	Increase in volume of interior of tube and longitudinal extension
Wertheim	Annales de Chimie et de Physique, ser. 3, vol. 23, 1848.	glass				
Kirchoff	Poggendorff's Annalen, vol. 108, 1859.	steel		0.294	Wire subjected to combined twisting and bending	Twist and transverse deflection
Bauschinger	Der Civilingenieur, 1879	steel		0.284	Bar subjected to tension	Direct measurements of cross contraction and longitudinal extension
Bauschinger	Der Civilingenieur, 1879	steel (bessemer car axle)		0.261		
Bauschinger	Der Civilingenieur, 1879	tin		0.275		
Bauschinger	Der Civilingenieur, 1879	sandstone		0.075 to 0.250		
Amagat	Journal de Physique, 1889	steel	29.60×10^6	0.269	*a* Tube closed at both ends subjected to longitudinal tension	*a* Increase in volume of interior of tube and longitudinal extension
Amagat	Journal de Physique, 1889	brass	15.73×10^6	0.327	*b* Tube closed at both ends subjected to (1) longitudinal tension and (2) to external pressure over its surface	*b* (1) Increase in volume of interior of tube (2) decrease of volume of interior of tube
Amagat	Journal de Physique, 1889	copper	17.82×10^6	0.327		
Amagat	Journal de Physique, 1889	lead	2.28×10^6	0.428		
Mallock	Proceedings Royal Society, vol. 29, 1879	steel		0.253	Rectangular beam bent by couples	Longitudinal and transverse radii of curvature of surface

Mallock	Proceedings Royal Society, vol. 29, 1879	brass		0.325		
Mallock	Proceedings Royal Society, vol. 29, 1879	copper		0.348		
Morrow	Philosophical Magazine, ser. 6, vol. 6, 1903	steel (mild)	30.64×10^6	0.275	Bars subjected to tension	Direct measurement of cross contractions and longitudinal extension
Morrow	Philosophical Magazine, ser. 6, vol. 6, 1903	steel (Sheffield spindle)	29.05×10^6	0.276		
Morrow	Philosophical Magazine, ser. 6, vol. 6, 1903	brass (drawn)	13.70×10^6	0.341		
Morrow	Philosophical Magazine, ser. 6, vol. 6, 1903	copper (drawn)	17.73×10^6	0.327		
Schellens	Thesis M. I. T. 1911	steel (mild structural)	29.97×10^6	0.282	Bars jacketed with mercury	Increase in volume of mercury jacket and longitudinal extension
Schellens	Thesis M. I. T. 1911	steel (high carbon tool)	29.43×10^6	0.292		
Howard	Tests of Metals, Watertown Arsenal	steel				

by far the most powerful gyroscope plant ever undertaken, he had two 50-in. wheels weighing on the order of 5000 lb. each, upon which rupturing tests, so called, were run. The real object of the tests was (*a*) to determine the integrity of the specific strength of the wheels; (*b*) to check them up with the elastic limit as indicated by a number of test bars taken from wheels; and (*c*), the most important, to make the interior portion of these wheels flow in such a way that in their natural state the whole interior of the wheel would be under pressure before finished boring. The speed was calculated at which the central part of the bore of the wheels, the general shape of the hubs of which were similar to Fig. 2 of the paper, would take a permanent set to the extent of several thousandths of an inch. The wheels were then put to this speed and the bore searched carefully for evidences of the permanent set. In each case it was found that this not only existed but was practically in accord with the quantities calculated. This was also an assurance that the wheels were solid and reliable in this critical zone or central region. Experience with these wheels would corroborate the statements made in the paper, which is an extremely valuable and reliable contribution to the knowledge on the subject.

The Author. As Mr. Rice has already stated, Professor Kenerson's point in regard to the validity of the measurements at high speeds was covered by the calibrating sleeve tests discussed in Par. 22. As stated, this sleeve had two different bore diameters, each being accurately known. The sleeve was a thin ring so that its increase of bore at full speed was very small and could be accurately computed. The amount was 0.0012 in. The increase of the bore shown by the instrument agreed exactly with this, showing that there was no compression of the pistons by centrifugal force, indentations of the wheel bore by the piston ends or other unlooked for effects at high speeds. The decrease in length of the pistons due to centrifugal force was computed and found to be absolutely insignificant. The ends of the pistons were rounded and carefully hardened.

In regard to Poisson's ratio there is but one published set of American experiments so far as I know; the matter has been completely settled by many European experiments, however. Through the kindness of Mr. C. A. Schellens I am able to append Table 2 giving the principal references in this matter.

No. 1371

INVESTIGATION OF EFFICIENCY OF WORM GEARING FOR AUTOMOBILE TRANSMISSION

By Wm. H. Kenerson, Providence, R. I.

Member of the Society

This investigation was made at the plant of the Brown & Sharpe Manufacturing Company for the purpose of determining the efficiency of three types of worm gearing for use in an automobile transmission system and the heating effect due to continuous running. The power for these tests was obtained from a 50-h.p. induction motor, running approximately 870 r.p.m. at full load.

2 Between the motor and the worm gear case was placed an automobile transmission gear case to enable tests to be made at two lower speeds. Between this and the worm gear case was placed a transmission dynamometer designed by the author and described in a paper presented before this Society and published in Vol. 31 of Transactions.[1] An Alden brake was used to absorb and measure the power transmitted by the gears under test. The arrangement of the apparatus is shown in Fig. 1 which shows the transmission dynamometer set up for comparison with the Alden brake, and in Fig. 2 which shows the whole apparatus mounted ready for testing the worm gear efficiency. Referring especially to Fig. 2, *A* is the motor, *B* the automobile transmission gear case, *C* the transmission dynamometer, *D* the case containing the worm gear under test, *E* the Alden brake, and *F* the platform scale which measured the load on the Alden brake.

3 The apparatus was so arranged that as the load was imposed the weight on the platform scale was removed and all vibration of the scale beam was eliminated by interposing springs *S* between the blocks *FF* which sustained the weight on the

[1] Wm. H. Kenerson, A New Transmission Dynamometer, p. 171.

Presented at the Annual Meeting 1912, of The American Society of Mechanical Engineers.

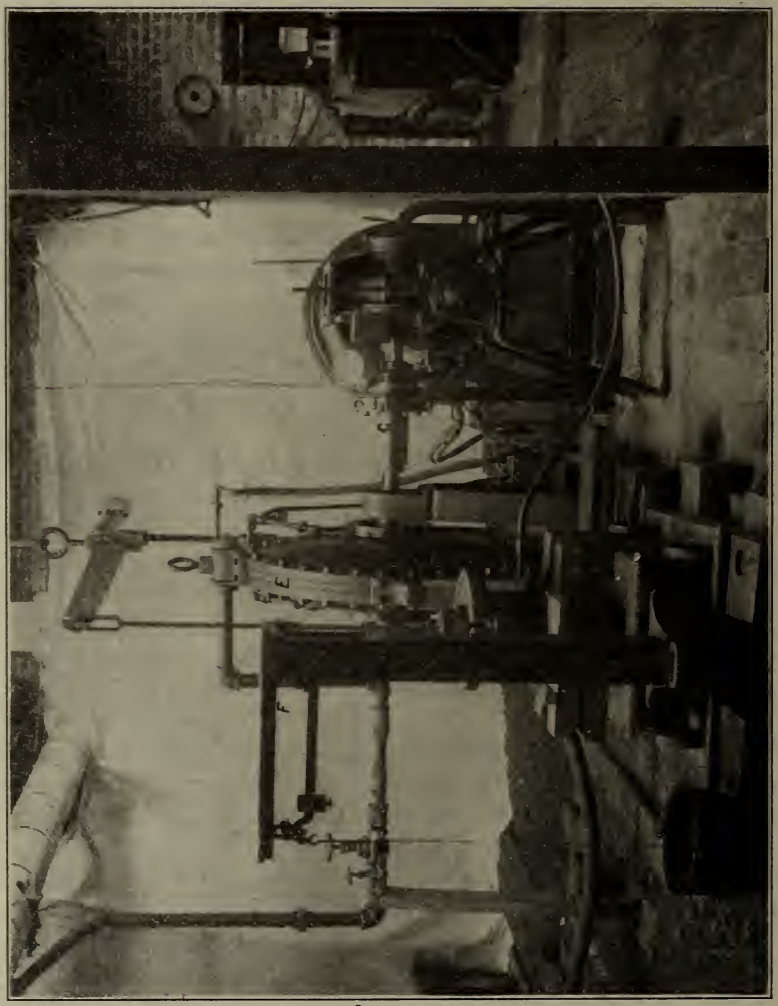


FIG. 1 APPARATUS ARRANGED FOR CALIBRATION

FIG. 2 APPARATUS FOR TESTING WORM GEARS

scale, and also by suspending by a wire from the weight at the end of the scale beam a plate which dipped into a pail of oil, thus acting as an efficient dashpot. It was found possible by careful manipulation to maintain a steady and easily read load on both the transmission and absorption instruments.

4 The transmission dynamometer and brake were first compared by arranging them as shown in Fig. 1, in which position

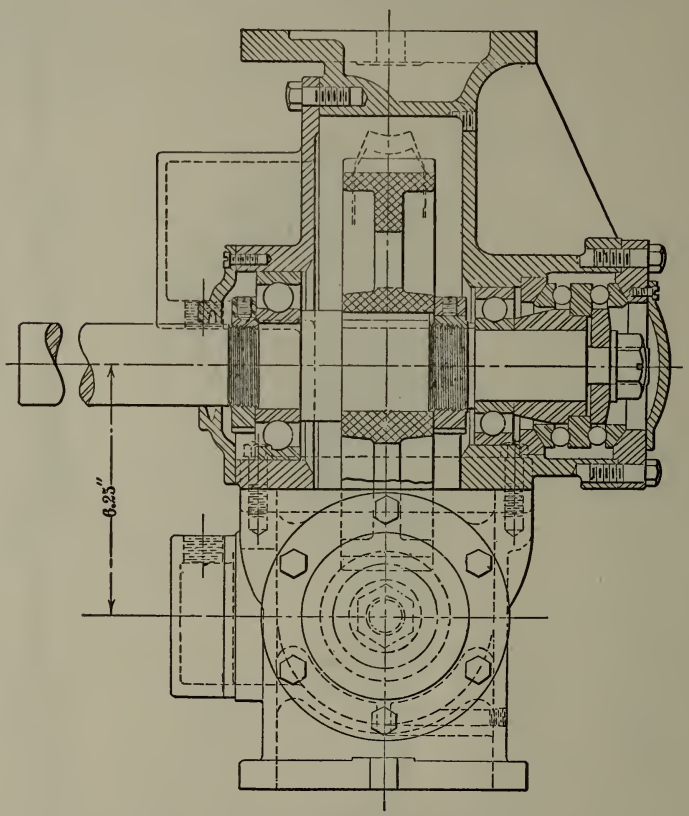


FIG. 3 SECTION THROUGH GEAR CASE

runs were made at various loads, and the torques corresponding to horsepower per 100 r.p.m. marked on the dial of the transmission instrument corresponding to similar loads as indicated on the brake. It is evident that by this method of comparison the two instruments must check each other exactly.

5 A thermometer placed in the oil well at the back of the

worm gear case *D* indicated the temperature of the oil, and another thermometer placed on the wall near the apparatus indicated the room temperature.

6 The oil employed to lubricate the gears was one intended for use with superheated steam, having a specific gravity of 26 Baumé, flash point 625 fahr., and viscosity at 212 deg. 260 to 265. The case contained about 5 quarts of the oil.

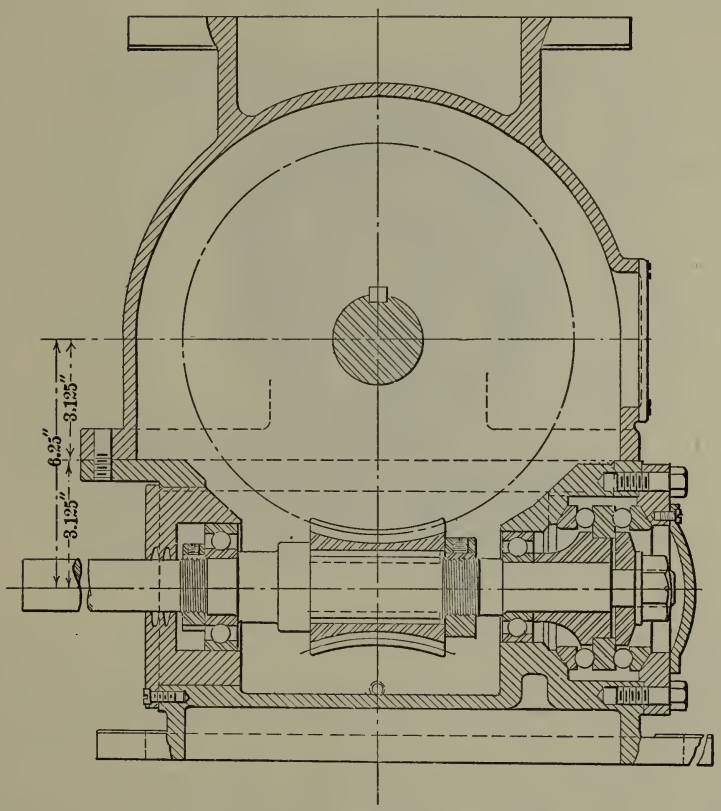


FIG. 4 SECTION THROUGH GEAR CASE

7 In all the trials the worm was located underneath the gear. Figs. 3 and 4 show sections through the gear case. As indicated, both shafts are mounted on ball bearings and end-thrust ball bearings take care of the thrust on the worm and wormwheel. All the worms were made of machinery steel, case-hardened, and the wormwheels of phosphor bronze.

8 The first worm and wheel tested are shown in Fig. 5, and are similar to those used for driving the spindle on the Brown & Sharpe automatic spur gear machines. This pair of gears is an unusual case of the worm and wormwheel. The smaller gear is hobbed with a hob of the size of the larger gear, thus making possible adjustment of the larger gear, which would not otherwise be the case. While in appearance this gear resembles a Hindley worm, it is not such in reality.

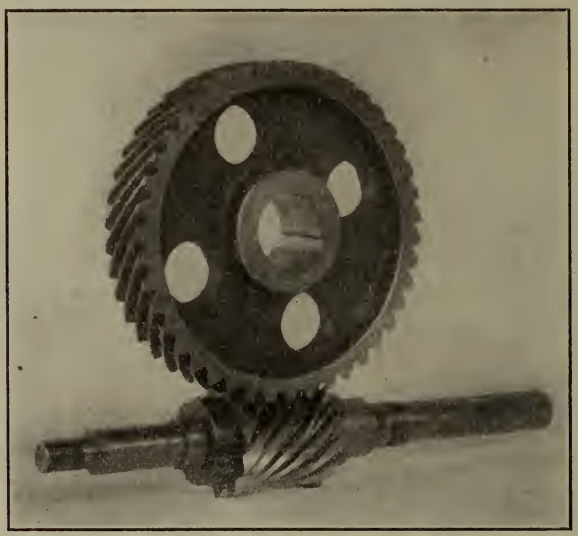


FIG. 5 FIRST WORM AND WHEEL TESTED

9 Although not strictly in accordance with usage, the smaller gear will be called the worm and the larger the wormwheel in the following description. Figs. 5 and 6 give the dimensions and a good general idea of this pair. The wormwheel was cut with the cutter shown, and the shape of the teeth on a section through the worm and wormwheel, parallel to the axis of the worm, is also shown.

10 All necessary information pertaining to pairs Nos. 2 and 3 is given in Figs. 7, 8 and 9. The difference between gears 2 and 3 is principally one of shape of the worm threads. This difference is clearly brought out in the sections of the worm and wheel shown in Figs. 10 and 11.

11 The shape of the teeth on the worm in Fig. 10 was produced with a cutter, the included angle of which was 29 deg., and the depth of tooth 0.570 in. This depth was based on the axial pitch, when the usual method on multiple worms is to base the depth on the normal pitch. The object in using this cutter was to obtain as many teeth as possible in contact at one time, and also a shape that could be ground with a straight-sided emery wheel.

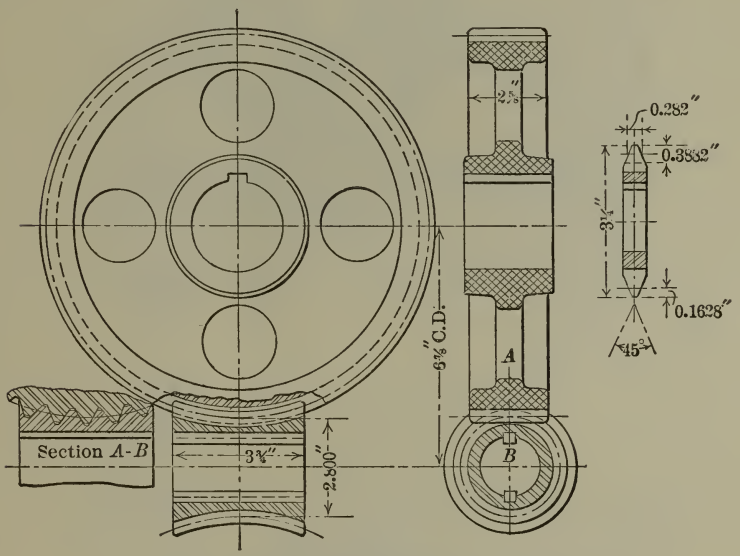


FIG. 6 NO. 1 WORM AND WORMWHEEL

Wormwheel, Phosphor Bronze. No. of teeth, 43 L. H.; Pitch diameter, 10.95; Outside diameter, 11.28; Circular pitch, 0.800; Angle of teeth with axis, 45 deg.; Normal circular pitch, 0.5657; Pitch of cutter, 5.553; Addendum, 0.1628 (not standard); Thickness of tooth, 0.282; Whole depth, 0.3882; Included angle of cutter, 45 deg.

12 The teeth of the worm in Fig. 11 were cut with a cutter shaped as shown, which is an arbitrary shape made to produce at one instant the greatest effective breadth possible, as shown in the figure.

13 In conducting the trials the load was maintained at the desired point by one observer who adjusted the brake. Readings were then taken on the transmission instrument by two independent observers. The speed of the motor was observed in each

case and from this, knowing the gear ratio, all the other speeds were easily computed. Temperatures were taken immediately following each series of observations.

14 For purposes of comparison a series of trials was run on a pair of bevel gears, the principal dimensions of which are shown in Fig. 12. Table 1 is a record of one set of observations typical of the series. Table 2 is a summary of the results of all

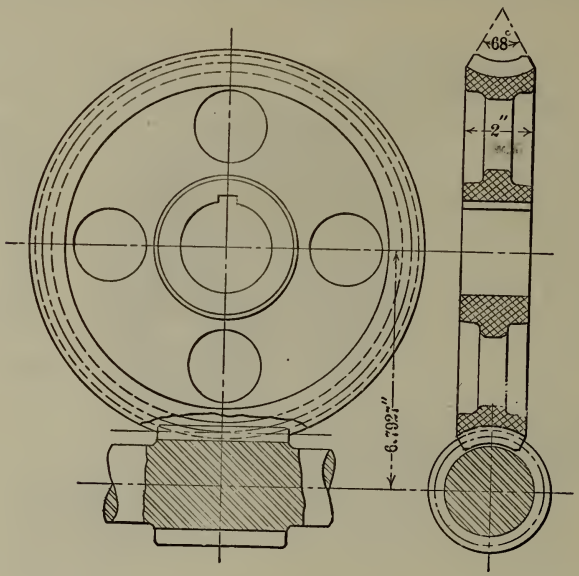


FIG. 7 NOS. 2 AND 3 WORM AND WORMWHEEL

Wormwheel, phosphor bronze; Number of teeth, 40; Pitch diameter, 10.5704; Throat diameter, 10.9964; Circular pitch, 0.8302; Angle of teeth with axis, 38° 16′ 5″; Normal circular pitch, 0.6518; Pitch of cutter, 4.8196; Addendum, 0.213; Thickness of tooth, 0.3568; Whole depth, 0.4586; Worm: Aurora Steel, Case-Hardened; Number of teeth, 9; Pitch diameter, 3.015; Outside diameter, 3.441; Circular pitch, 1.0524; Angle of teeth with axis, 51° 43′ 55″; Thickness of tooth, 0.295; Lead, 7.4719; Ratio of Wheel to Worm, 40 to 9.

the trials, and the curves of Fig. 13 show in graphic form the average efficiency of the different gears at the various loads and speeds.

15 In conjunction with the efficiency trials a series of runs was made to determine the heating effect due to continuous running. In these trials, which were in effect endurance tests, a

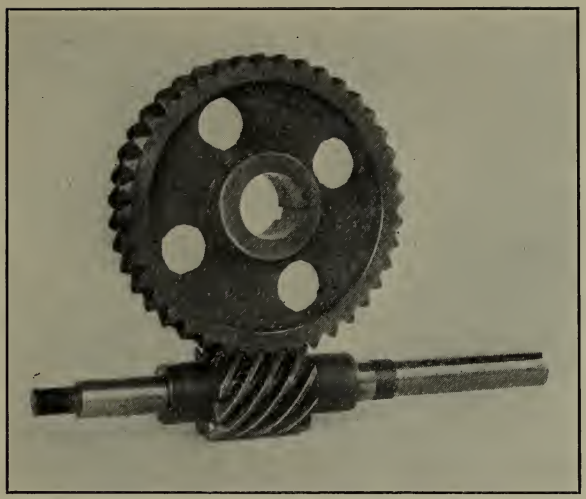


FIG. 8 No. 2 WORM AND WORMWHEEL

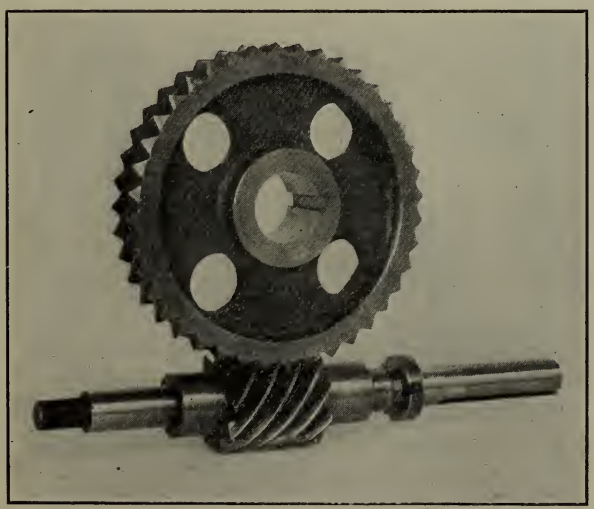


FIG. 9 No. 3 WORM AND WORMWHEEL

TABLE 1 EFFICIENCY TEST OF NO. 2 WORM GEARS WHEN ON DIRECT SPEED

R.P.M. Motor	R.P.M. Worm	R.P.M. Worm Gear	Speed of Worm in Ft. per Min.	Initial Load	Reading on Scale	Load on Brake 315 In. Radius	H.P. per 100 R.P.M. Brake	Total Brake H.P.	Total Trans. Dyn. H.P.	Efficiency of Worm Gears	Transmission Dyn. Reading				Temp. at End of Test		Time		Duration
											1	2	3	Av.	Oil	Room	Start	Fin.	
875	875	196.9	690.7	104	55	49	24.5	48.24	49.26	97.9	5.6	5.65	5.65	5.63	268	65	10:14	10:20	6
873	873	196.4	689.1	104	60	44	22.0	43.22	44.35	97.5	5.1	5.05	5.1	5.08	273	65	10:20	10:25	5
880	880	198.0	694.6	104	65	39	19.5	38.61	40.92	94.4	4.7	4.65	4.6	4.65	276	66	10:25	10:30	5
880	880	198.0	694.6	104	70	34	17.0	33.66	36.26	92.8	4.15	4.1	4.1	4.12	275	66	10:30	10:33	3
883	883	198.7	697.0	104	75	29	14.5	28.81	31.17	92.4	3.55	3.55	3.5	3.53	273	66	10:33	10:37	4

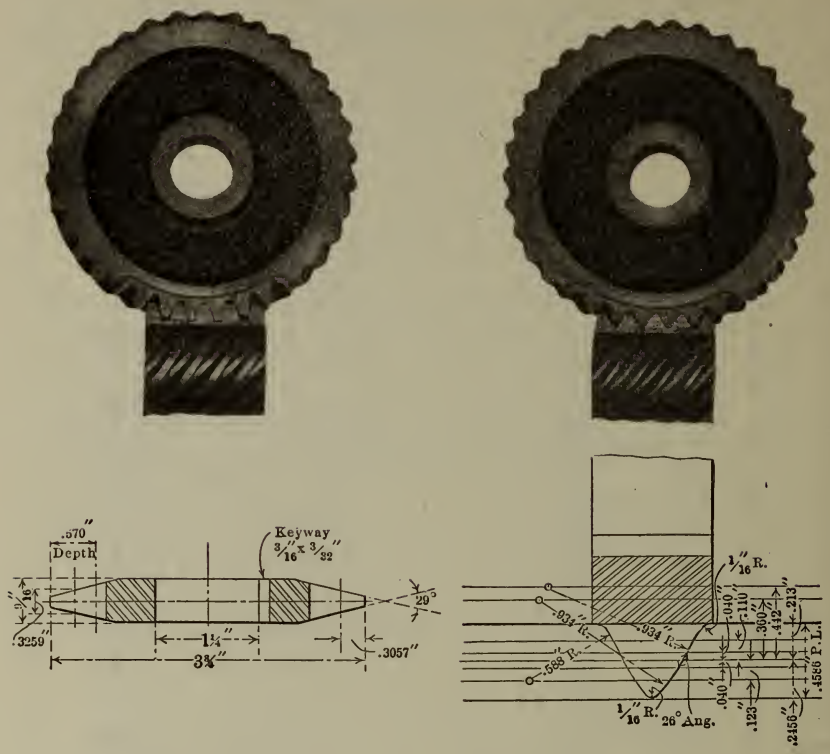
NUMBER OF GROUPS OF READINGS TAKEN IN TABLE 2

	Gear No. 1	Gear No. 2	Gear No. 3	Bevel Gear
First Speed....	4	5	5	3
Second Speed..	4	5	5	6
Direct.....	3	5	5	5

Total Separate Readings, 165

TABLE 2 WORMS AND WHEELS AND BEVEL GEARS, AVERAGE EFFICIENCIES TAKEN

	Gear No. 1							Gear No. 2							Gear No. 3							Bevel Gears						
	R.P.M.		H.P.		Temp.		Eff.	R.P.M.		H.P.		Temp.		Eff.	R.P.M.		H.P.		Temp.		Eff.	R.P.M.		H.P.		Temp.		Eff.
	Worm	Worm Wheel	Input	Output	Oil	Room		Worm	Worm Wheel	Input	Output	Oil	Room		Worm	Worm Wheel	Input	Output	Oil	Room		Pinion	Gear	Input	Output	Oil.	Room	
First Speed	254.2	65.03	26.18	24.06	206	73	91.9	254.7	57.32	19.26	18.34	164	69	95.3	254.1	57.19	18.75	18.30	136	79	95.5	254.2	68.43	20.92	20.19		70	96.6
	254.6	65.13	24.75	22.47	217	73	90.8	254.7	57.32	17.99	16.91	169	69	94.1	254.8	57.33	16.54	15.48	144	79	93.1	254.5	68.51	17.50	16.78		70	95.9
	254.5	65.11	22.98	20.81	225	73	90.5	254.5	57.25	16.73	15.46	162	68	92.5	255.0	57.38	13.75	12.62	149	79	91.8	256.1	68.95	14.54	13.44		69	92.4
	255.1	65.25	21.17	19.25	228	73	90.9	254.3	57.22	15.08	14.02	142	69	93.1	255.4	57.47	10.80	9.77	150	79	90.5	256.5	69.05	10.95	10.01		70	91.4
	255.3	65.30	19.56	17.63	232	73	90.1	255.1	57.39	13.52	12.63	159	69	93.5	255.6	57.52	7.38	6.90	149	79	93.6							
	255.7	65.41	17.76	16.02	233	74	90.2	254.9	57.35	12.14	11.18	160	69	92.2														
	255.9	65.48	16.00	14.41	233	74	90.1	254.8	57.34	10.69	9.75	160	69	91.4														
	256.3	65.60	14.34	12.79	233	74	89.4																					
	256.5	65.62	12.37	11.15	231	74	90.2																					
	256.8	65.69	10.55	9.53	229	74	90.2																					
Second Speed	545.2	139.5	43.48	41.16	178	73	94.8	544.2	122.4	36.95	36.11	178	70	97.7	543.2	122.2	38.59	36.06	204	77	93.4	541.8	145.0	39.64	39.37		67	99.3
	546.9	139.9	40.54	37.78	207	70	93.0	545.0	122.6	34.61	33.10	184	70	95.6	543.8	122.4	35.36	33.04	215	77	92.4	544.1	146.4	33.43	32.22		67	96.4
	547.8	140.1	36.88	34.33	214	70	93.2	546.7	123.0	32.09	30.14	188	70	93.9	544.8	122.6	33.12	30.03	219	77	90.7	545.7	146.9	26.25	24.98		67	95.2
	548.8	140.4	33.47	30.89	217	70	92.3	548.4	123.4	28.90	27.15	187	70	93.9	545.8	122.8	29.82	27.02	221	77	90.6	548.5	147.6	18.75	17.72		67	94.5
	550.2	140.7	29.76	27.44	218	70	92.3	549.0	123.5	25.80	24.08	186	71	93.3	546.5	123.0	26.70	23.98	222	78	89.8							
	550.5	140.8	26.01	23.94	218	71	91.3	549.6	123.7	22.59	21.03	163	70	93.1	547.2	123.2	23.56	20.95	222	78	88.9							
	551.1	141.0	22.22	20.45	217	71	92.0	551.0	124.0	19.12	17.98	163	70	94.0	548.3	123.4	20.02	17.89	221	78	89.4							
Direct	859.5	219.8	66.43	64.86	178	69	97.6	873.5	196.5	49.35	48.14	265	66	97.5	866.0	194.8	49.95	47.72	215	77	95.5	872.0	234.7	52.05	51.63		70	99.2
	864.5	221.1	62.89	59.71	201	70	94.9	875.0	196.9	44.50	43.30	256	65	97.3	868.2	195.3	45.68	42.98	223	77	94.1	878.6	236.6	42.09	40.22		70	95.5
	871.0	222.8	59.23	54.60	214	70	92.2	878.0	197.5	40.40	38.50	260	66	95.2	870.8	195.9	41.20	38.20	228	77	92.7	884.4	238.2	30.40	28.58	132	79	94.0
	874.5	223.7	54.22	49.22	224	70	90.8	881.0	198.2	36.40	33.70	261	66	92.4	874.0	196.7	36.88	33.45	230	77	90.6							
	878.0	224.6	48.60	43.80	230	70	90.1	883.4	198.8	30.73	28.84	262	66	93.8	876.0	197.1	31.45	28.60	230	78	90.9							
	882.0	225.6	42.25	38.36	233	70	90.8								878.8	197.7	26.45	23.73	229	78	89.7							
	884.0	226.1	36.24	32.79	237	71	90.5																					



FIGS. 10 AND 11 SECTION OF NOS. 2 AND 3 WORM AND WORMWHEEL

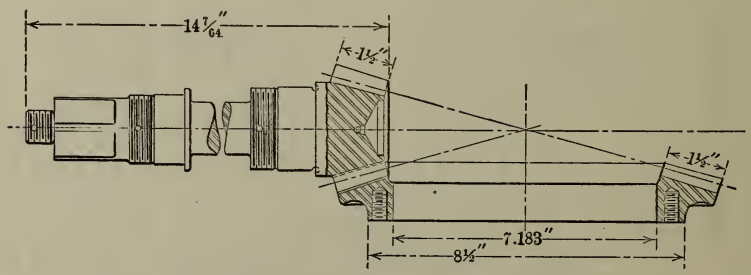


FIG. 12 DRIVING GEAR AND PINION

Driving Pinion, 5 Per Cent Nickel Steel, Case-Hardened; Pitch, 5; Number of teeth, 14; Angle of edge, 15° 4′; Angle of face, 71° 5′; Outside diameter, 3.3359. Driving gear, 5 per cent nickel steel, case-hardened; Pitch, 5; Number of teeth, 52; Angle of edge, 74° 56′; Angle of face, 13° 47′; Outside diameter, 10.4627.

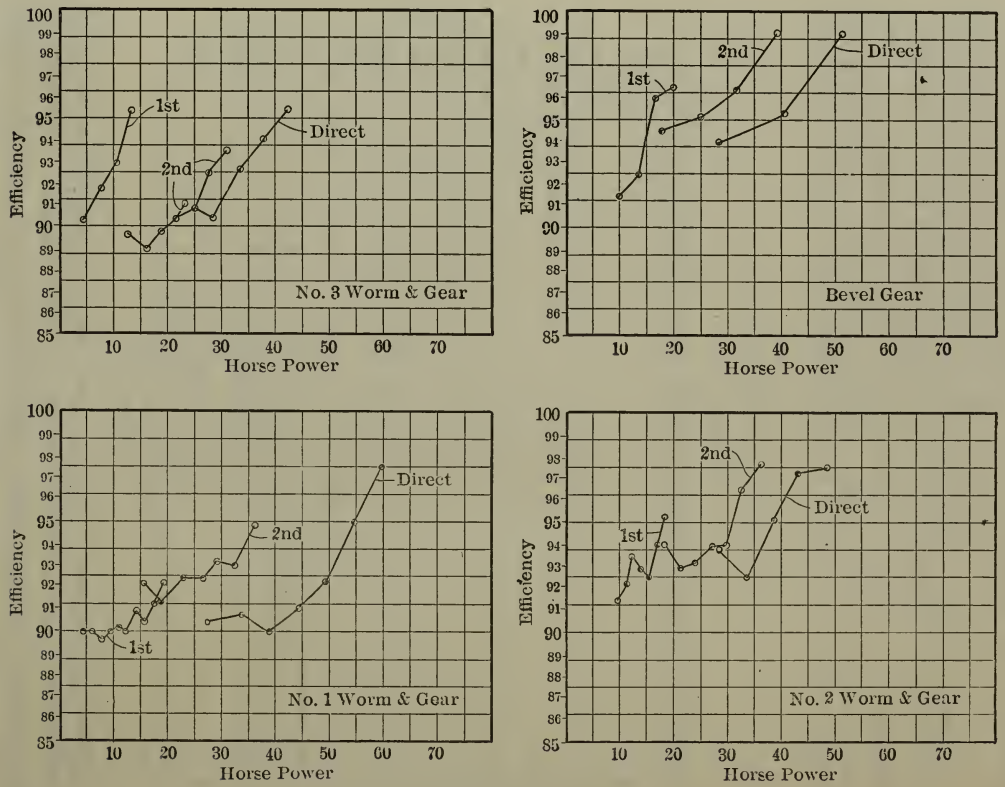


FIG. 13 EFFICIENCY CURVES

constant load was transmitted through the gearing and the temperature of the oil in the gear case and the temperature of the room noted at frequent intervals.

16 From these observations it was found that at the beginning of the run the oil temperature rose rapidly and somewhat irregularly. As the run continued, however, the rise became much more gradual and regular. In the runs where the smaller

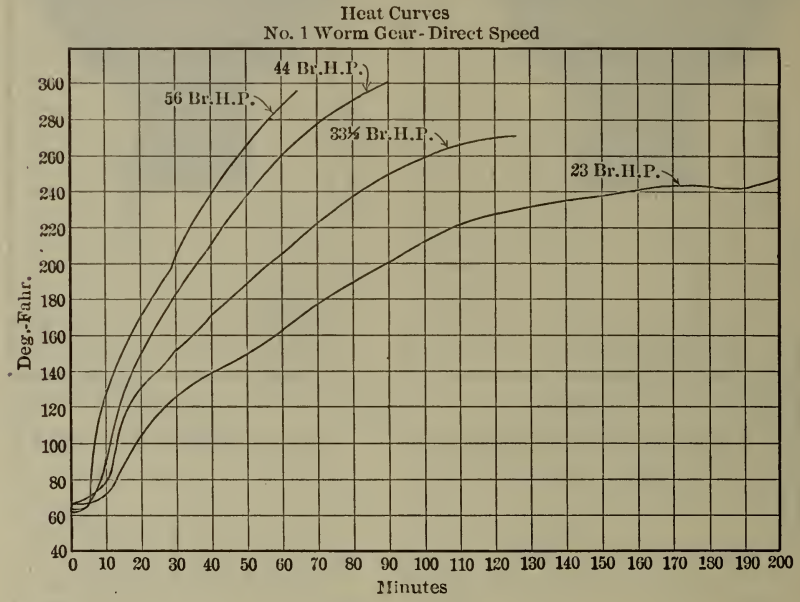


FIG. 14 HEAT CURVES

amounts of power were transmitted a point was reached where the temperature remained constant. This indicated that radiation was sufficient to carry away the heat due to power lost through friction in the gearing, or in other words that the gears would run indefinitely at the load. The heat curves of the No. 1 worm and gear are shown in Fig. 14.

17 The higher loads indicated were abnormal for the gears under consideration and would not occur in any use to which the gears would normally be put. The fact that these trials continued for from 60 to 80 minutes without failure indicates that the structure is both strong and enduring and that should such temperatures be reached for any accidental reason the gears

would not be destroyed. The result of the trials was of particular interest because of the very high efficiency and carrying capacity of the gears tested.

18 Every possible precaution was taken to secure accuracy in the results, and the high degree of accuracy obtained is due largely to the skill and care of Mr. B. F. Waterman of the Brown & Sharpe Manufacturing Company, under whose personal supervision the apparatus was erected and all the trials were conducted.

DISCUSSION

Elmer A. Sperry. Question was raised as to the correctness of the evidence contained in the paper that the coefficient of friction decreases with increase of pressure. Within the past year the writer has conducted some interesting experiments on two large gyroscopes with oil bearings and running up to about 50 ft. per sec. with loads varying through a range of about 3000 per cent upon the lower half of the bearing and more than this on the upper half. Upon the lower half of the bearing the actual load ranged approximately from 2200 lb. to 65,000 lb., whereas on the upper half of the bearings the load varied from zero to 60,000 lb. This pressure could be applied at will, practically rising and falling in the form of a sine curve between the minimum and maximum.

The coefficient at gravity load was about 0.008, and for the highest load about 0.0036 to 0.004, as nearly as it could be determined. The writer's experience has invariably been that as the pressure would increase the coefficient would come down under the conditions named, the apparent range being somewhat in excess of 100 per cent. The results being entirely concordant and dealing as they did with large quantities, they may be considered as reliable and possibly contributing further to our knowledge on this extremely important subject.

Regarding Mr. Kent's remarks on the influence of the area of contact upon the coefficient of friction, is it not possible that the efficiency would be quite independent of the area actually involved? For instance, in the journals above referred to, which were ordinary oil bearings provided with excellent oil circulation employed for the twofold reason of lubrication and cooling, it was observed that after 160,000 precessions of these gyros, involving very heavy pressures on the journals, there was an

entire lack of evidence of metallic contact, or that only a small portion of the entire area had ever been involved in sustaining these loads. For instance, one photographic record shows that only about 20 per cent of the projected area had ever been employed in the actual work as the original tool marks are present over the remaining surfaces.

W. C. Marshall. I would ask why the efficiency increases with the increase of load and speed, and why it is greater on the first and second speeds than on the direct drive. In the case of first and second speeds other gears are introduced and the efficiency should be less. In some cases it is greater with other gears introduced than with the direct drive. The rubbing velocity at lower speeds is less, but Mr. H. Kerr-Thomas of the Pierce-Arrow Company definitely states that in all his tests the efficiency does not vary appreciably between low and high speed with varying loads. The efficiency depends on the frictional resistance which varies with the square root of the velocity, and inversely as the pressure on the teeth. With an increase of pressure on the teeth and the velocity uniform there would be a reduced coefficient of friction. But if the velocity is increased and the pressure uniform the numerator of the frictional coefficient will be increased and consequently the resistance. If both quantities are varied (the denominator and the numerator), the result will remain nearly constant. Of course, this does not apply to extremely high speeds.

In the author's tests, however, the efficiency decreases in several cases with increase in load and speed, which I do not understand. It seems to point to the destructive coefficient which comes into play at high speeds and large loads. This coefficient is the product of the pressure and the velocity and the tendency is for it to cause heating. This is shown in these experiments, as the oil becomes thinner and allows metallic contact, which reduces the efficiency.

It seems to me that the coefficient of friction in these experiments is rather high for a large load, as indicated by the rising temperature of the oil, and I have made one or two calculations to find out what this coefficient was, taking the figures given in the first line of Table 1. From the specific heat of the oil, the weight of the oil, and the number of heat units given to this oil in 6 minutes, I find 131 B.t.u. were given to the oil in 1 minute.

Substituting that in the expression $H = \frac{p\mu V60}{J}$ for heat units in terms of the pressure, coefficient of friction, velocity and Joules' equivalent, the coefficient of friction comes out as 0.033. Working with that coefficient, which is rather large, the efficiency of these gears would figure out as 93 per cent $E = \frac{1 - \mu \cot \alpha}{1 + \mu \tan \alpha}$. It did not figure out anywhere near 93 per cent in this case. If Unwin's recommendation for the coefficient of friction = 0.002 is used in such cases, the efficiency will be in the neighborhood of 0.996. Mr. Thomas uses the same one in making calculations on worm gears, in which case the efficiency comes out at 0.996 per cent, which is a great deal higher than any of the experiments shown in the paper. The angle α or the helix angle is 38 deg.

Bevel gears afford such a high efficiency that it does not seem at all advantageous to use worm gears in driving automobiles unless the noise is a consideration. I would like to know if there was noise in connection with the test on bevel gears. That is an important point, especially if there is any resonant quality to the support.

In a test made on worm gears by Frank Burgess a curve of efficiencies is given which, using small h.p., runs almost in a straight line from 1 h.p. up to 3 h.p., where the efficiency is 93 per cent, then rises in the case of 4 h.p. to 95 per cent, and then begins to drop at a steady rate to 91 per cent at 8 h.p. Professor Kenerson's experiments do not show anything like this. There is no drop; always a rise. But there must be a point where the drop takes place, and I would like to see some results to show it. The automobile transmission is important and it should be ascertained where the efficiency decreases. Also there must be high efficiency at low speed, since nine-tenths of the time automobiles are running under reduced power, and if the efficiency is very low it does not pay to use worm gearing.

Mr. Kerr-Thomas has proved that under certain conditions, such as trucking, the efficiency is just as high at low speeds as at high. Professor Kenerson does not agree with him at all, and I would like to get an expression of opinion from other members of the Society in regard to this point.

William Kent said he would like Professor Kenerson to give some theoretical reason why at uniform speed of rotation and increasing pressures, and consequently increased horsepower, the efficiency increases. It might be possible in starting on low speeds where the worm gears were imperfectly polished, that they were not as smooth as they should be and that the viscosity of the oil was greater at the low temperatures, but it is improbable that either of these causes would be sufficient to account for the lower efficiency at low pressures.

In regard to calculating the coefficient of friction, Mr. Kent saw no way that it could be done, because it was impossible to calculate the area of the surface that is subjected to pressure, that is, the area of contact.

L. B. Taylor. Mr. Kent called attention to the fact that the coefficient of friction has usually been considered to increase with the increasing specific pressure, while the experiments in question indicate the reverse.

The laws governing copiously lubricated surfaces, as determined by Petroff and given in the Hütte, 1911 edition, Vol. 1, Page 244, are as follows:

The coefficient of friction varies

a directly as the coefficient of the internal friction of the oil used at the temperature prevailing

b directly as the relative velocity of the rubbing surfaces.

On the other hand the coefficient varies

c inversely as the actual mean thickness of the layer of oil between the rubbing surfaces

d inversely as the specific pressure exerted between the contact surfaces

e at constant temperature the thickness of the layer of oil is inversely proportional to the square root of the specific loading of the surfaces in contact.

Since these laws were discovered the specific pressure allowed on steam turbine bearings has been increased 50 per cent.

The Author. Referring to Professor Marshall's remarks, his understanding that the use of speed gears will affect the efficiency is incorrect. The dynamometer is placed between the change gears and the gear under test.

In comparing the results of the Burgess test with those under

consideration, it should be borne in mind that the gears were tested under very light loads. Mr. Kerr-Thomas' statement that under certain conditions, such as trucking, efficiency is as high at low speeds as at high, undoubtedly refers not alone to the worm gearing but to the whole machine.

This statement is undoubtedly true because of the fact that the power lost in the tires of an automobile is much in excess of that lost in the transmission, and, of course, decreases with the speed. The question raised by both Professors Marshall and Kent regarding the decrease in coefficient of friction with the increased load is referred to and answered by others in the discussion.

There are so many factors to be considered that I believe it impossible to calculate a coefficient of friction as suggested by Professor Marshall.

Professor Kent's question regarding the reason for increased efficiency with the load is answered by the fact that the friction does not increase directly with the horsepower.

No. 1372

CASE CARBONIZING

By Marcus T. Lothrop, Canton, Ohio

Junior Member of the Society

During the last decade the better understanding of the constitution of ferrous alloys and the improvement in apparatus have led to great advances in the commercial heat treatment of steel. The progressive manufacturer today carefully inspects the steels as received and before he allows them to enter his shop knows their composition and their physical condition. After the raw material is fabricated into finished parts, these parts are heat treated to show the desired physical properties. This operation of heat treatment must be conducted with certainty and uniformity and an inspection of existing conditions indicates that the uniformity is much greater when the manufacturer does not change the composition of the metal being heat treated. Case carbonizing, however, requires that the carbon content of the material on the surface be increased while the original composition of the material in the interior is retained. The operation requires that the user, in a sense, make his own steel and the details of the operation must be understood to the end that the operation may be conducted with the minimum cost and with the maximum certainty and that variations and irregularities shall not creep into the product.

2 The investigations indicate that, commercially, the case-carbonizing operation can be performed in such a manner that the final result will be absolutely uniform and that the advantages which accrue from case carbonizing represent the most efficient manner of obtaining the desired physical properties in many constructions. It was with the object of pointing out the causes of and the results effected by irregularity and the means

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and precautions which must be employed to produce uniformity, that these researches were undertaken.

3 The following points were studied:

a The effect of the composition of the steel on the rate and depth of carbonization.

b The effect of heat treatment on case-carbonized steel.

c The effect of time on the rate and depth of carbonization.

d The effect of temperature and of composition of the steel on the depth and nature of carbonization.

e The efficiency of various commercial carbonizing materials.

f The effect of depth of case carbonizing on the physical properties of the finished materials.

g The commercial thermal conductivity of the various case-carbonizing materials.

h The efficiency of copper plating in preventing carbonization.

The materials used are summarized in Table 1.

EXPERIMENT NO. 1

THE EFFECT OF HEAT TREATMENT IN CASE-CARBONIZING STEELS (TABLES 12-21)

4 The determination of a test method which would indicate the physical properties of the metal tested required considerable trial. A case-carbonized piece of metal, consisting of a hard exterior and a soft and tough interior, is so dual in its physical characteristics that tensile tests are not adequate. This is due to the fact that when the case which is hard and brittle fails, the pull is suddenly applied to the reduced section of the core and causes it to fail so quickly that the expected ductility of the material fails to appear.

5 The test finally decided upon was a transverse test which has the advantage of approaching the condition of stress generally encountered in service. The test piece adopted was ½ in. round, 12 in. long, with 10 in. between supports, loaded at the center. A round section gives a case-carbonized zone of uniform depth in all parts and for this reason is superior to the flat section which has excessive depth in the corners. The 10-in. center to center distance between supports allows loads large

enough to be measured accurately and gives a deflection at rupture great enough to be easily measured. By means of this test, pieces of the same material and heat treated in identically the same way give results for strength and toughness which check closer than by any other method tried.

HEAT TREATMENT METHOD

6 These test pieces were packed in wrought-iron pots 23 in. long by 5 in. round, with 0.25 in. walls, so that each was 1 in.

TABLE 1 MATERIALS USED

No.		C	Si	Mn	P	S	Cr	Ni	V
2	0.15% C basic open hearth.....	0.15	0.02	0.68	0.054	0.043	....	....	....
3	0.20% C basic open hearth.....	0.20	0.02	0.66	0.038	0.044	....	....	....
4	0.25% C basic open hearth.....	0.25	0.04	0.61	0.038	0.046	....	....	....
5	0.10% C Halcomb electric furnace nickel steel.....	0.10	0.13	0.66	0.010	0.025	....	3.60	....
6	0.15% C Halcomb electric furnace nickel steel.....	0.15	0.09	0.70	0.009	0.013	....	3.54	....
	0.20% C Halcomb electric furnace nickel steel.....	0.20	0.18	0.69	0.007	0.013	....	3.59	....
8	0.25% C Halcomb electric furnace nickel steel.....	0.25	0.21	0.74	0.009	0.009	....	3.55	....
9	0.31% C Halcomb electric furnace nickel steel.....	0.31	0.18	0.70	0.009	0.010	....	3.70	....
10	0.20% C Halcomb electric furnace chrome vanadium steel.....	0.20	0.13	0.50	0.009	0.010	1.00	....	0.15
11	0.50% C Halcomb electric furnace chrome vanadium steel.....	0.49	0.15	0.80	0.009	0.021	1.18	....	0.19
12	0.50% C Halcomb electric furnace chrome nickel steel.....	0.48	0.16	0.44	0.009	0.009	0.98	2.02	....

from the inside wall of the pot and 1 in. from each other. Case-carbonizing compound No. 1[1] was used in this experiment. The precaution of having everything clean and dry and the pots well and uniformly tamped down was carefully observed, and identical conditions of rate of heating, cooling and temperature maintained. The temperature was 1700 deg. fahr. with a variation of not greater than plus or minus 15. The test pieces after case carbonizing were cleaned and reheated in a neutral atmosphere to the temperatures given in Tables 12-21, Appendix No. 1, which are correct to plus or minus 10 deg. fahr. Table 2 sum-

[1] The analyses of these and other compounds are given in Table 7

TABLE 2 INCREASES DUE TO HEAT TREATMENT

No.	Increase in Strength with Double Heat Treatment, Per Cent	Increase in Toughness with Double Heat Treatment, Per Cent	Drawing Single Treatment Increase		Drawing Double Treatment Increase		Annealing Single Treatment Increase		Annealing Double Treatment Decrease	
			Strength, Per Cent	Toughness, Per Cent	Strength, Per Cent	Toughness, Per Cent	Strength, Per Cent	Toughness, Per Cent	Strength, Per Cent	Toughness, Per Cent
2	23.0	105.0	12.0	25.0	....	....	10.0	26.0	....	....
3	25.0	56.0	21.0	4.0	....	....	16.0	4.0	....	....
4	12.0	80.0	12.9	18.2	....	....	14.3	18.4	....	....
5	20.0	10.8	24.8	10.8	9.1	20.0	2.4	6.5	....	....
6	6.1	....	23.0	17.8	11.0	10.0	4.1	....	....	....
7	5.6	13.6	27.4	20.5	14.5	16.0	4.0	....	1.5	6.0
8	6.9	19.0	29.6	43.0	17.0	23.8	1.5	2.4	0.7	8.0
9	6.9	....	34.4	27.6	25.0	40.5	4.1	2.1	3.3	....
10	10.7	92.5	14.3	25.0	16.7	13.6	2.8	7.0	....	....
Average carbon steel....	20.0	80.0	15.0	16.0	....	....	13.0	16.0	....	....
Average nickel steel.....	8.0	14.0	27.6	24.0	15.3	20.0	3.6	2.0	1.1	3.0

marizes the results due to reheating and annealing, showing the increases in strength and toughness. Table 3 gives the temperatures at which the case ceases to be file hard as a result of drawing. All the tests recorded in the tables are the average of closely agreeing duplicates, and were made on test pieces which after heat treatment were file hard.

GENERAL CONCLUSIONS, EXPERIMENT NO. 1

7 A study of the tables leads to the following conclusions:

a Strength and toughness decrease with increasing depth of case. This is important commercially for with a certain steel and a certain heat treatment uniformity of strength and toughness is impossible if the depth of case varies.

TABLE 3 EFFECT OF DRAWING IN CASE-CARBONIZED STEELS

No.	Steel No. and Grade	Temperature at which Case Ceases to be Glass Hard, deg. fahr.
1	1—.10 per cent—C—Bessemer	425
2	2—.15 per cent—C—O. H.	425
3	3—.20 per cent—C—O. H.	425
4	4—.25 per cent—C—O. H.	425
5	5—.10 per cent—C—E. F. Ni.	425
6	6—.15 per cent—C—E. F. Ni.	425
7	7—.20 per cent—C—E. F. Ni.	425
8	8—.25 per cent—C—E. F. Ni.	425
9	9—.30 per cent—C—E. F. Ni.	425
10	10—.20 per cent—C—E. F. Cr. V.	450

The temperatures are such that the case can just be touched with a new fine file.

b The critical depth of case at which maximum brittleness and minimum strength occur, depends upon the initial carbon content of the steel which is case carbonized. The higher the original carbon content, the smaller the ratio of depth of case to diameter of core at the point where this brittleness occurs. The commercial importance of this point is that the manufacturer using the steel for case carbonizing should know its composition and should vary the depth of the carbonizing effect as demanded by the composition of the metal being used.

c Strength and toughness increase with double heat treat-

ment. This result is to be expected, for the dual nature of case-carbonized steel requires two thermal treatments properly to refine the composite metal.

d Strength and toughness increase when the temper is drawn at 380 deg. fahr. This operation does not decrease the hardness but relieves in a greater or less degree the strains set up in hardening and makes for uniformity as well as more pronounced physical properties.

e Annealing after case carbonizing is a doubtful operation. The physical improvement with single heat treatment may be obtained by resorting to double heat treatment, for the results will be better and the time consumed in the operation much shorter. Annealing does not improve the double heat treated pieces and may even cause loss of strength and toughness, due to the decarbonization which may take place. Annealing is not to be recommended.

f The case-carbonized steel will, with either double or single heat treatment, become file hard at temperatures too low for the best development of the maximum strength and toughness. Although seemingly paradoxical, the dual nature of the steel after case carbonizing will explain this established fact, when one recalls that the transformations of the exterior are first completed at low temperatures, and that those of the interior are last completed at higher temperatures.

g All steels lose their file hardness when drawn at 425 deg. fahr., save chrome vanadium which loses it at 450 deg. fahr.

h The ideal heat treatment for a case-carbonized steel may be assumed to be: Case carbonize to the thinnest possible depth of case demanded by the conditions of service; reheat for the core; quench in a suitable fluid; reheat for the case; quench in a suitable fluid; draw the temper as far as the conditions of service will permit.

i Case-carbonized steel parts, such as gears, etc., may be stronger, tougher and harder than similar parts made from oil-hardened steels.

EXPERIMENT NO. 2

MICROSCOPIC EXPERIMENT ON THE EFFECT OF HEAT TREATMENT AFTER CASE CARBONIZING

8 The object of this experiment was to determine, if possible, the constitutional difference between a case-carbonized bar of steel which had received a single heat treatment (reheating being a compromise, in that the temperature selected may be a little too high for the best refinement of the carbonized zone, and a little too low for the best refinement of the core), and a piece of the same metal carbonized in the same manner which had received a double heat treatment, the first reheating designed to be the best possible for the interior or the core and the second reheating designed to be the best possible for the case or carbonized zone.

9 Steels Nos. 3, 7 and 10 were case carbonized at 1300, 1400, 1500, 1600 and 1800 deg. fahr., and were treated with the best single and double heat treatments as detailed in Experiment No. 1. The specimens were etched for 10 seconds in a 1 per cent nitric acid solution. All specimens were martensitic with double heat treatment.

a The martensite was easily distinguished in steel No. 3 (0.20 per cent carbon open hearth) at 100 diameters magnification.

b The martensite was distinguished with difficulty in the No. 7 steel (0.20 per cent carbon electric furnace nickel steel) at 100 diameters magnification.

c The martensite was not distinguishable in steel No. 10 (0.20 per cent carbon chrome vanadium steel) at 100 diameters magnification. Martensite was barely distinguishable in this steel at 400 diameters magnification.

10 With single heat treatment

a The martensite was very easily distinguished in the No. 3 steel at 100 diameters magnification.

b The martensite was easily distinguished in the No. 7 steel at 100 diameters magnification.

c The martensite was just barely distinguishable in the No. 10 steel at 100 diameters magnification. At 400 diameters the martensite with this steel was easily distinguished.

11 The results of this experiment indicate that the double heat treatment increases the strength and toughness of case-carbonized steels due to the smaller and finer martensitic structure developed.

EXPERIMENT NO. 3

THE EFFECT OF TEMPERATURE USED IN CASE CARBONIZING

12 The object of this experiment was to determine the lowest temperature at which case carbonizing begins; the lowest temperature with each of the three different types of case-carbonizing steels (straight carbon and alloy, all of the same carbon content) at which it might be most efficiently conducted in practice; and the best temperature for the case carbonizing of steel.

13 The following steels were used:

a No. 3 0.20 per cent carbon open-hearth carbon steel
b No. 7 0.20 per cent carbon electric nickel steel
c No. 10 0.20 per cent carbon electric chrome vanadium steel

14 The test pieces were ½ in. round and 6 in. long, properly numbered for identification; three, one of each grade of steel, were packed in a pot for each run. The pot used in this experiment was made of wrought iron 4 in. round and 8 in. long with ¼-in. wall. A solid steel plug was welded into one end, and the other end was fitted with a removable sheet-steel cover which was drilled in the middle with a hole large enough to accommodate the electro-thermo-couple protection tube. The test pieces were placed in the pot, equally spaced about the electro-quartz protection tube in the center and packed tightly in case-carbonizing compound No. 1. All heating was done in a gas-fired muffle furnace. Temperature readings of the furnace temperature and the inside temperature of the pots were taken every 15 minutes. Standardized platinum platinum-rhodium thermo-couples were used. Six runs of 5 hours each at a definite temperature were made, using the same apparatus and procedure. The six runs were at the following temperatures in deg. fahr.:

No. 1.....1300	No. 3.....1500	No. 5.....1700
No. 2.....1400	No. 4.....1600	No. 6.....1800

15 The furnace was started cold in each case and time was counted from the moment the pot couple showed a temperature

inside the pot corresponding to the temperature for which the run was intended. After this, the pot temperature was maintained as close as possible to this point for an uninterrupted period of 5 hours, at the end of which time the pot couple was removed, the pots pulled from the furnace and allowed to become cold.

16 The temperature records of this series of experiments are given in Tables 22-27 of Appendix No. 2. Following these tables in the same appendix is a detailed analysis of the microscopic examinations of the various test pieces after being subjected to the heat treatment and specimen microphotographs are reproduced in Figs. 18 to 30, shown full size.

17 The conclusions to be drawn from this analysis follow herewith, together with a tabulated statement (Table 4) of the thickness, in millimeters, of the zone of excess cementite, of the eutectic, of the gradation zone and the total penetration of carbon in millimeters for the different steels.

CONCLUSIONS FROM MICROSCOPIC INVESTIGATION OF OPEN-HEARTH STEELS CASE CARBONIZED AT 1300 TO 1800 DEG. FAHR.

18 In the specimen treated at 1300 deg. fahr. a small zone of carbonization was clearly separated from the interior material having a great number of penetrating bands containing the so-called nitrides in very minute needles. In the gradation zone these nitrides appeared in large quantities in larger needles.

19 In the specimen treated at 1400 deg. fahr the carbonized zone was slightly thicker and the gradation zone larger, giving a more gradual division between the carbonized and interior material. The penetrating bands were present as before in large numbers. Nitrides were still present inside the eutectic zone, but not in such great numbers as in the first specimen.

20 In the specimen treated at 1500 deg. fahr. the carbon penetration was slightly deeper, and a zone of excess cementite was found on the outside, which appeared as large irregular areas and as polygonal outlines. No nitride needles of any kind were visible.

21 The specimens treated at 1600 deg. fahr. and 1700 deg. fahr. showed still deeper penetration of carbon, greater excess of cementite, and a larger number of cementite needles.

22 At 1800 deg. fahr. there was a much greater penetration of carbon with an outer zone of excess cementite, but this cement-

ite did not occur in polygonal outlines, except in a few places. The general appearance of the carbonized zone was that of a eutectic, containing areas of excess cementite, grading to the normal carbon content of the interior.

CONCLUSIONS FROM MICROSCOPIC INVESTIGATIONS OF NICKEL STEELS CASE CARBONIZED AT 1300 TO 1800 DEG. FAHR.

23 In this series there was a slight carbonization in the speci-

TABLE 4 SUMMARY CASE-CARBONIZING EXPERIMENT NO. 3[1]

	Excess Cementite			Thickness of Eutectic		
Temperature. Deg. Fahr.	O. H.	Ni	Cr Va	O. H.	Ni	Cr Va
1100	.....	.....	.....	.....	.....	.....
1200	.....	.....	.....	.....	.....	.....
1300	0.008	.....	trace	0.090	.....	.....
1400	0.090	.....	0.009	0.120	0.203	0.030
1500	0.128	0.056	0.285	0.150	0.278	0.135
1600	0.135	0.158	0.375	0.210	0.390	0.180
1700	0.225	0.293	0.465	0.338	0.488	0.248
1800	0.293	0.390	0.338	0.450	0.390	0.405
	Gradation Zone			Total Penetration of Carbon		
	O. H.	Ni	Cr Va	O. H.	Ni	Cr Va
1100	.....	.....	.....	.....	.....	.....
1200	.....	.....	.....	.....	.....	.....
1300	0.034	0.180	0.016	0.140	0.030	0.180
1400	0.188	0.390	0.300	0.410	0.590	0.420
1500	0.420	0.510	0.338	0.710	0.830	0.770
1600	0.450	0.683	0.278	0.770	0.990	0.890
1700	0.600	0.975	0.450	1.160	1.560	1.370
1800	0.683	0.900	0.675	1.640	2.030	1.750

[1] The measurements in this table give the width of zones in millimeters.

men treated at 1300 deg. fahr. only, and no penetration bands similar to those of the open-hearth series which contained nitrides.

24 In the specimen treated at 1400 deg. fahr. there was a better penetration of carbon, but no higher carbon content on the outside than eutectic. Several of the penetrating bands were present.

25 At 1500 deg. fahr. there was a still greater penetration of carbon, but little if any excess cementite on the outside, and few of the penetration bands were present in this specimen.

26 In the 1600 deg. fahr. specimen there was a normally increasing depth of case, with a slight amount of excess cementite on the outside and bands of penetrating material.

27 In the specimen treated at 1700 deg. fahr. there was still greater penetration of carbon and an increase in the excess cementite on the outside of the carbonized zone. In this specimen there were also a few penetrating bands. The cementite existed as crystal boundaries and as needles in the crystal cleavages.

28 In the specimen treated at 1800 deg. fahr. a great deal of excess cementite was found on the outside, as well as greater penetration of carbon and no penetrating bands. This series differed from the open-hearth series in that the excess cementite and the excess ferrite in their respective zones did not tend so uniformly to form the polygonal outlines as they did in the open-hearth series. Also either no nitrides were present in this series, or, if they were in the penetrating bands, they were in a different form from any nitrides so far observed.

CONCLUSIONS FROM MICROSCOPIC INVESTIGATION OF CHROME VANADIUM STEELS CASE CARBONIZED AT 1300 TO 1800 DEG. FAHR.

29 In this series, the specimen treated at 1300 deg. fahr. showed a small zone of carbon, the highest carbon content on the outside being about eutectic.

30 In the specimen treated at 1400 deg. fahr. there was a very slight rim of cementite and pearlite in about equal amounts, while at 1500 deg. fahr. the excess cementite was found in two distinct zones, the outer zone containing the cementite in large globules intermingled with pearlite, the inner zone of polygonal outlines of cementite surrounding pearlite areas.

31 At 1600 deg. fahr. the cementite appeared in two zones as at 1500 deg. fahr.

32 At 1700 deg. fahr. the double zone appearance of the cementite was not so marked, and at 1800 deg. fahr. only one zone contained the cementite in polygonal outlines. The polygons as a rule appeared to be larger towards the inner part of the zone.

GENERAL SUMMARY OF THE EFFECT OF TEMPERATURE IN CASE CARBONIZING

33 The following is a general summary of the effect of temperature in case carbonizing:

a The depth of case carbonizing effect in a given time increases with the temperature.

b The carbon content in the case-carbonized zone in a given time increases with the temperature. This affords the commercial opportunity of varying the depth of case-carbonizing effect as desired by changing the time, and of varying the carbon content of the case-carbonized zone by changing the case-carbonizing temperature. It is poor practice to raise the temperature of the case-carbonizing operation for the purpose of reducing the time which the operation consumes, for this will increase the maximum carbon content of the case-carbonized zone and may change the character of the physical properties of the finished product.

c The minimum temperature at which uniform penetration of carbon can be obtained with case carbonizing seems to be 1500 deg. fahr. With lower temperatures the carbon maximum is not uniform in its distribution and the penetration too slow.

d The presence of excess cementite in the form of needles within the pearlite grains, so conspicuous in nickel and carbon steels, indicates a temperature during case carbonizing which overheats the metal being so treated.

e Nickel steel gives the greatest total penetration and greatest gradation zones.

f Chrome vanadium gives the highest carbon maximum and finest grained steel.

g When case carbonizing is conducted at 1400 deg. fahr. or lower, the carbon maximum does not exceed the eutectic. As the case-carbonizing temperature increases, a zone of hypereutectic composition develops which increases in depth with the temperature. The depth of this zone becomes important when sufficient warpage takes place in the piece being hard-

ened to demand grinding after the final heat treatment. It has been demonstrated often that a carbon content of more than 1 per cent in the surface of a case-carbonized part is undesirable when there is the least shock to be reckoned with, because in such instances the excess of carbon exists in the form of films of carbide between the grains of the metal which are brittle and hard, weakening the steel and rendering it liable to fail by flaking off or spawling. Therefore it would be inadvisable to select a practice or a material which produces such a condition in the surface. On the other hand if the carbon percentage is lower than 0.90 to 1.00 per cent the case will not show its maximum hardness. Therefore, when pieces are to remain unground practice should aim at a production of a 0.90 per cent carbon surface but when a portion of the steel is to be ground off, after the heat treatment, it is advisable to drive in an excess of carbon so as to leave this optimum percentage of 0.9 to 1.00 per cent in the final surface.

h All carbonized zones widen with increase of temperature. While wide gradation zones are desirable from the point of view of tenacity of the case to the core, other considerations enter into the commercial side of the problem. The first is the cost of maintaining the high heats. The factors of fuel cost and increased furnace up-keep cost at the upper temperatures are self-evident. The second is the difference in the nature of the case. Inspection will show that at 1500 deg. fahr. the width of the eutectic zone is considerably greater than that of the hypereutectic zone and the hypoeutectic zone of the decreasing carbon content is wider still, which means, of course, that the case carbonizing is proceeding gradually and that the case and core are merging into each other by gentle degrees. At 1600 deg. fahr. the ratio of the high carbon exterior zone to the inner zones rises and at 1700 and 1800 deg. fahr. the increase is more marked still.

The meaning of all this is that as the temperature goes up carbon is forced by the heat, as it were, to rush into the surface of the steel faster than it can be

assimilated by the adjacent grains and so the outside layers at the high heats are gorged with carbon which has not time to diffuse gradually toward the center of the case. This condition of affairs is highly undesirable, because the product then shows a sharp demarcation between case and core, the transition from one material to another being too abrupt; the properties necessarily vary accordingly and in the finished piece have a tendency to set up high physical strains. This causes flaking off of the case from the core, a phenomenon familiar enough to those engaged in the trade. The remedy is to case carbonize at a moderate heat for a length of time sufficient to obtain a reasonable depth of case with gradual transition from case to core. This temperature appears in this experiment to be about 1650 deg. fahr. with open-hearth carbon steel, 1600 deg. fahr. with electric nickel steel, and 1650 deg. fahr. with electric chrome vanadium steel.

In deciding upon the best carbonizing temperature, the effect of heat on the size of grain in the steel must also be considered. This as is well-known, is a very decided reaction, the grain coarsening rapidly with increasing temperatures. In this connection a careful investigation showed marked contrast between the behavior of ordinary carbon steel and the two alloy steels; the former began to coarsen rapidly at 1500 deg. fahr.; nickel steel also showed the same effect at the same temperature but in a much less marked degree and chrome vanadium steel retained its fine grain up to 1700 deg. fahr. and over, not reaching even at 1800 deg. fahr., the grain size of other steels. This is in line with the common belief that chromium and vanadium retain fineness of grain in steel.

EXPERIMENT NO. 4

THE EFFECT OF THE COMPOSITION OF THE STEEL BEING CASE CARBONIZED ON THE RATE AND DEPTH OF CARBONIZATION

34 The object of this experiment was to determine by chemical analysis the depth of carbonizing effect and carbon content of several zones of case carbonizing effect as affected by the composition of the steel being case carbonized.

35 Test bars 8 in. long and 1½ in. round were prepared by machining on centers in a lathe and packed in case-carbonizing compound No. 1, care being taken to see that all bars were evenly spaced. The bars were case carbonized at 1700 deg. fahr. for 7 hours. The pot was then removed from the furnace and

TABLE 5 EFFECT OF COMPOSITION OF STEEL IN CASE COMPOSITION AFTER CARBONIZING

No.	Material	Percentage of Carbon in Cuts of Various Depths, In.					
		0 to 0.005	0.005 to 0.010	0.010 to 0.015	0.015 to 0.020	0.020 to 0.025	0.025 to 0.030
1	No. 1 0.10% C Bessemer	0.90	0.63	0.32	0.16	0.13	0.11
2	No. 3 0.20% C O. H.	0.95	0.78	0.47	0.27	0.20	0.20
3	No. 6 0.15% C E. F. Ni	0.90	0.65	0.44	0.28	0.19	0.15
4	No. 9 0.30% C E. F. Ni	0.95	0.75	0.55	0.34	0.30	0.30
5	No. 10 0.20% C Cr Va	1.22	0.78	0.49	0.30	0.20	0.20

allowed to become cold, when it was unpacked and the bars removed and cleaned. The bars were set up in a lathe and cuts 0.005 in. deep taken. The sample thus obtained was analyzed by combustion and in duplicate for carbon, and the results will be found in Table 5. The information obtained from this experiment indicates that:

a The higher the carbon content of the steel case carbonized, the higher will be the maximum carbon content of case when case carbonizing is performed at a given temperature for a given time.

b The original carbon content does not affect the depth of case under the conditions of this experiment.

c The nickel and carbon steels behave similarly in regard to carbon maximum.

d Chrome vanadium steel takes a higher carbon maximum than nickel steel and carbon steel under the same case-carbonizing conditions.

EXPERIMENT NO. 5

EFFICIENCY OF CASE-CARBONIZING MATERIALS

36 This experiment was undertaken to determine the commercial efficiency of case-carbonizing materials and to show the elements which affect their value commercially.

37 Fourteen case-carbonizing materials were selected, representing the most prominent compounds used in this country. Fifty-pound samples were obtained on the open market from the several manufacturers, with prices, general description, etc. As soon as received average samples of about 1 pint each were taken, put in stoppered bottles, and sent to the chemical laboratory for analysis and specific gravity determination. These compounds are designated as Nos. 1, 2, 3, etc., in Table 4. The bars used in making up the test pieces (Table 1) were selected by chemical analysis and were all of 0.20 per cent carbon.

38 The test pieces were of three different types:

a Solid cylinders 6 in. long by 1.344 in. in diameter. These were cut to length and turned from 1⅜ in. diameter hot rolled and annealed bars; set up on centers in lathe and rough turned true and straight and close to finish size; finished absolutely straight and to size by grinding on centers. This left the bars entirely bright and with an almost polished surface ready for carbonizing.

b Hollow cylinders 6 in. by 1¼ in. outside diameter by ¼ in. thick wall. These were also made from 1⅜ in. diameter, hot rolled and annealed bars, turned to size and bored; not ground.

c Solid rod ½ in. in diameter by 6 in. long. Samples were cut to size from ½ in. in diameter (cold drawn) bars and sand-blasted, and all test pieces properly numbered for identification. Carbonizing pots used were new and of cast iron, rectangular in shape, inside dimensions 10 in. by 6¼ in. deep, with ½ in. thick wall. They were supported on legs ¾ in. off

furnace floor and prevented from touching each other in furnace by a boss 1½ in. long on each end (Fig. 1).

39 The pots were as carefully and as nearly uniformly packed in each case as possible. Three test pieces, one of each type, were used in each pot and were spaced as indicated.

40 A layer of carbonizer about 1½ in. thick was put in first and tamped down, then the lowest bar with more carbonizer, etc., tamping in snugly throughout, and so on until the pot

TABLE 6 PHYSICAL AND PRICE DATA ON CARBONIZING MATERIALS

Carbonizing Material No.	Weight of 1 Pt., Grains	Approximate Specific Gravity	Weight of 1 Cu. Ft., Lb.	Volume of 1 Ton, Cu. Ft.	Cost per Cu. Ft.
1	450	0.954	59.4	33.7	$1.84
2	365	0.774	48.1	41.5	1.30
3	438	0.930	57.5	34.5	1.50
4	418	0.870	55.5	36.3	1.48
5	436	0.925	57.5	34.8	1.58
6	285	0.604	37.6	53.2	0.85
7	259	0.550	34.2	58.5	1.02
8	373	0.720	49.2	40.6	0.86
9	248	0.526	32.7	61.1	1.64
10	126	0.267	16.6	120.4	....
11	363	0.770	47.8	41.8	1.31
12	448	0.950	59.1	33.8	....
13	287	0.608	37.8	52.9	1.32
14	298	0.630	39.3	50.9	1.77

was full. The cast-iron covers were luted on with clay, small "tell-tales" inserted in one end, and pots put into the furnace.

41 The original weight of test pieces and carbonizing material used in each pot were obtained before each run.

42 The American Gas Furnace Company's gas-fired muffle furnace No. 4 was used in case carbonizing. It was found possible to run six pots at a heat, equally spaced in muffle, not touching each other or the sides of furnace. Furnace-working temperatures were obtained by regular Le Chatelier type thermocouple, the hot end of which projected down between the pots in the middle of the muffle. Each of the six pots used in a run contained a different kind of carbonizing agent. Three consecutive runs were made on each kind, the material from the preceding run being used in each case except, of course, the

first run, which was on new raw material. New test pieces were used for each run of each kind of carbonizer.

43 In conducting the carbonizing runs, the furnace working conditions were maintained as nearly uniform in each as possible. Pots were arranged in the muffle in the afternoon ready for starting and at 7 a.m. the furnace was lighted. The temperature was worked up at a fairly rapid rate and very close to 9 a.m. it had reached 1750 deg. fahr. in all cases. This temperature was maintained uniformly throughout the day and up to 6 p.m. when the furnace was shut off and furnace and pots were allowed to cool slowly over night. Subsequent examination included:

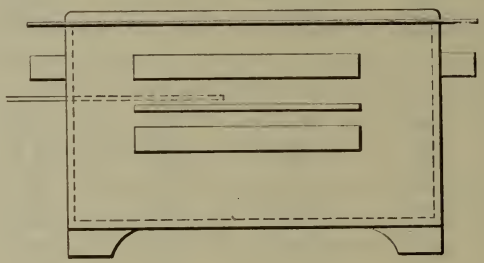

FIG. 1 DIAGRAM SHOWING LOCATION OF TEST PIECES IN CARBONIZING POT

a Thorough examination of general appearance of pots, carbonizing material and test pieces after unpacking; weights obtained to show loss of weight of carbonizer, or gain of weight of test pieces, loss of volume of carbonizer. The mixtures were carefully saved and used without further rejuvenation, in packing the pots for the next run. Shrinkage in volume was taken care of by using sufficient sand to fill up pots. This was done by placing the added sand in the bottom of the pot.

b Taking of cuts from solid cylinders for analysis. Bars after carbonizing, were carefully cleaned of all dirt, etc., set up on centers and tested for straightness or straightened, if necessary, until they ran absolutely true on centers. Then 11 consecutive cuts of 0.005 in. each on radius were taken, the length of the bar and turnings carefully kept separate and put in en-

TABLE 7 ANALYSIS OF CARBONIZING MATERIALS

Number of Material	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Price per Ton, Dollars and Cents	62	55	52	53.60	....	45	60	35	100	...	55	...	70	90
Approximate Analysis, Per Cent														
Moisture	5.65	5.57	6.50	5.90	7.03	2.4	1.70	2.26	4.60	4.10	1.55	0.45	6.15	0.85
Volatile Matter	31.17	33.73	32.90	33.14	30.87	25.77	10.73	12.43	10.90	15.95	32.35	23.10	54.95	15.53
Fixed Carbon	0.15	1.00	0.05	1.20	0.60	37.75	60.32	58.30	78.62	75.80	15.60	57.93	22.20	30.82
Ash	63.03	59.70	60.55	59.76	61.50	34.02	27.25	26.04	5.88	4.15	50.50	18.52	16.70	52.80
	100.00	100.00	100.00	100.00	100.00	99.94	100 00	99.03	100.00	100.00	100.00	100.00	100.00	100.00
Ash Analysis, Per Cent														
Calcium Oxide	34.31	32.27	not determined	not determined	33 49	17.00	7.95	6.64	0.52	0.67	26.17	8.39	6 68	....
Phosphorus Pentoxide	29.00	27.28			28.30	14.02	} 3.97	} 2 62	} 0.35	} 0.49	23.40	3.30	5.93	} 0.58
Aluminum Oxide	.....	.....			.....	} 1.82					} 0.76	} 0.52	} 0.28	
Iron Oxide	.....	.....			.....									
Silica	.....	.....			.....	0.58	5.76	5.46	0.11	0.08	0.34	0.79	0.69	1.58
Carbon Dioxide	.....	.....			.....	....	....	....	0.46	....	....	....	....	10.68
Sulphur Trioxide	.....	.....			.....	0.65	2.12	2.15	1.85	1.62	0.80	1.60	0.90	1.52
Sodium Oxide	.....	.....			.....	....	6.45 (by difference)	10.13 (by difference)	....	....	....	....	....	....
Magnesium Oxide	.....	.....			.....	....	....	....	1.43	1.18	....	3.29	0.63	0.68
Barium Oxide	.....	.....			.....	....	....	....	1.62	....	....	....	....	37.24
	100.28	99.85			100.29	100.05	99.00	100.00	100.46	99.89	100.97	99.37	98.41	99.48
Solid Matter and Carbon Dioxide, in the Carbonizer														
Total Combustible Carbon	12.51	14.07	15.86	14.88	13.50	53.42	66.24	65.88	78.41	77.66	41.42	70.11	40.02	44.93
Carbon Dioxide as Carbonates	3.51	2.95	} not determined	not determined	3.43	0.62	4.62	4.35	1.20	0.75	1.18	7.36	0.89	11.63
Total Sulphur	0.16	0.10		....	} not determined	0.26	1.03	0.97	0.74	0.65	0.24	0.64	0.36	0.61
Total Phosphorus	12.66	11.91		....		7.00	0.005	0.002	0.024	0.24	10.22	1.44	2.59	0.01
Sulphur as Sulphide	.....	.....	.....	....	.....	....	0.08	0.15	....	....	....	....	....	....
Hydro-Carbon Oils (soluble in gasoline)	.....	.....	.....	....	.....	....	....	....	....	....	23.82	....	....	....
Approximate Specific Gravity	0.954	0.774	0.930	0.88	0.92	0.60	0.79	0.55	0.52	0.27	0.77	0.95	0.61	0.63

TABLE 8 MEASUREMENTS IN MILLIMETERS OF ZONES AND CARBON PENETRATION OF TEST PIECES

See Appendix No. 2

Case-Carbonizing Material No.	Zone of Excess Cementite	Eutectic Zone	Gradation Zone	Total Penetration of Carbon
Run No. 1				
1	0.455	0.455	0.650	1.560
2	0.450	0.650	0.748	1.848
3	.....	0.520	1.235	1.755
4	.....	0.975	1.300	2.275
5	0.780	0.520	0.975	2.275
6	.....	.....	1.235	1.235
7	0.325	0.520	0.975	1.920
8	0.455	0.520	0.975	1.950
9	1.235	0.423	0.780	2.438
10	.....	0.975	1.105	2.080
11	.....	0.195	1.300	1.495
12	.....	0.339	1.235	1.574
13	0.325	0.520	1.300	2.145
14	1.040	0.339	0.650	2.029
Run No. 2				
1	.....	0.715	1.001	1.716
2	.....	0.650	1.073	1.723
3	.....	0.520	1.300	1.820
4	.....	0.878	0.975	1.853
5	.....	0.780	0.975	1.755
6	.....	.....	0.975	0.975
7	.....	2.260	1.300	1.560
8	0.325	0.715	0.910	1.950
9	1.075	0.455	0.780	2.300
10	.....	0.065	1.170	1.235
11	.....	.....	1.658	1.658
12	.....	0.130	1.560	1.690
13	.....	0.520	1.170	1.690
14	.....	1.300	0.450	1.105
Run No. 3				
1	.....	0.358	1.138	1.496
2	.....	.....	0.975	0.975
3	.....	.....	0.325	0.325
4	.....	0.455	0.975	1.430
5	.....	0.225	1.300	1.525
6	.....	.....	0.040	0.040
7	.....	0.715	1.170	1.995
8	0.195	0.650	0.650	1.495
9	0.325	0.845	0.975	2.145
10	.....	.....	0.910*	0.975*
11	.....	.....	1.235†	1.242†
12	.....	0.007	1.560	1.567
13	.....	0.293	1.170	1.463
14	1.170	0.845	1.300	3.315

* On the outside of this specimen there was a rim of almost pure ferrite 0.065 mm. thick.
† On the outside of this specimen there was a rim of almost pure ferrite 0.007 mm. thick.

velopes for subsequent determinations of carbon, phosphorus and sulphur.

c Breaking of carbonized tube for general appearance and action of carbonizer in tubes, etc.

d Breaking of ½-in. round rod (some after heat treatment) for general appearance, depth of case, and microscopic examination.

44 In Table 6 are listed data concerning the 14 carbonizing materials and in Table 7 are given the analyses. In Figs. 2 to 15 are shown graphically the carbon contents of the carbonized sections of the test bars at successive depths, as explained in Par. 43*b*. The captions of these diagrams give also the phosphorous and sulphur contents in each case in the first cut of 0.005 in. and of the core, respectively.

45 In Appendix No. 3 is a detailed description of the results of the microscopic examination of the several bars and in Table 8 is summarized the measurements in millimeters, determined by the microscopic examination, of the zone of excess cementite, the eutectic zone, the gradation zone and the depth of carbon penetration.

46 In Table 9 are summarized physical and price data on the carbonizing materials in relation to the carbon penetration of the test bars.

47 The results of the experiment on the efficiency of case-carbonizing materials may be summarized as follows:

a The volume per ton of case-carbonizing materials varies. As these materials are used by volume, not by weight, this volume per ton must be considered in the economic purchase of this grade of material.

b All the compounds show shrinkage of volume and in most instances a loss of potency with continued service. This point is of greatest commercial interest since, if steel of variable carbon content is produced in the case-hardening process the manufacturer cannot hope for uniform results.

c With the compounds on the market, any desired maximum carbon content in the case can be obtained with a given temperature simply by varying the compounds used.

d When parts are produced which require grinding after

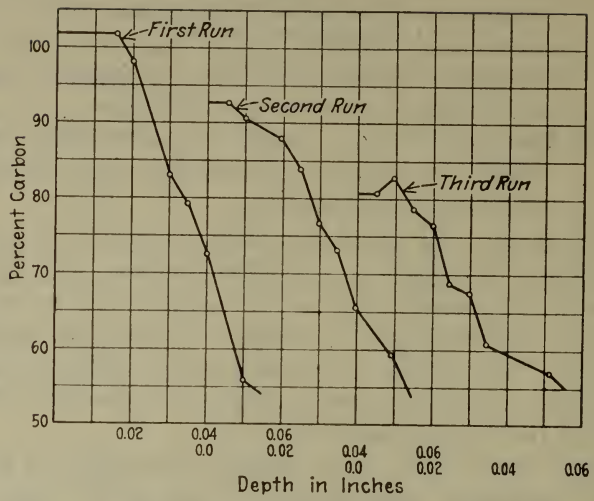

FIG. 2 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 1 (SEE PAR. 44)
PHOSPHORUS CONTENT 0.025%–0.029%; SULPHUR CONTENT 0.04%–0.04%

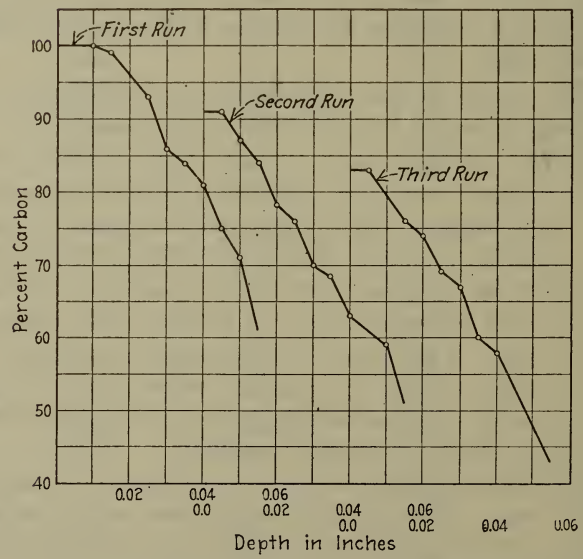

FIG. 3 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 2
PHOSPHORUS CONTENT 0.030%–0.032%

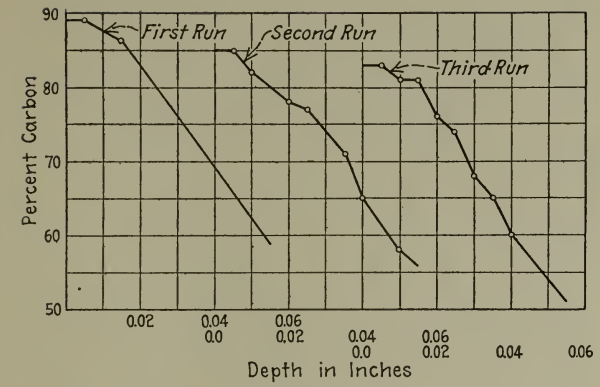

FIG. 4 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 3
PHOSPHORUS CONTENT 0.029%–0.033%

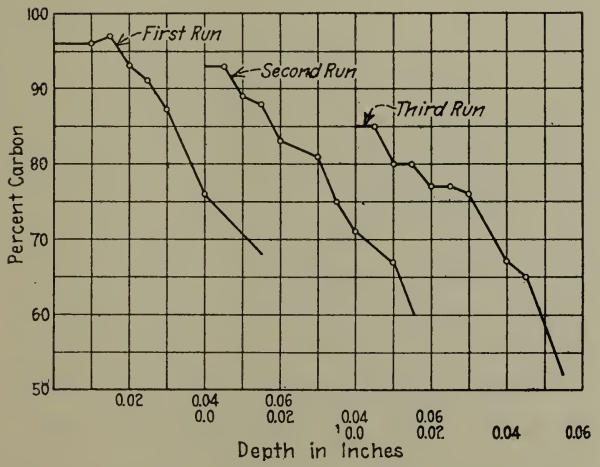

FIG. 5 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 4
PHOSPHORUS CONTENT 0.029%–0.036%

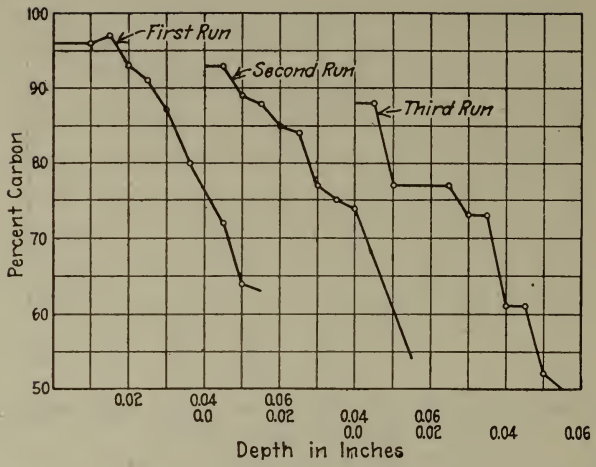

FIG. 6 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 5 (SEE PAR. 44)
PHOSPHORUS CONTENT 0.032%–0.038%

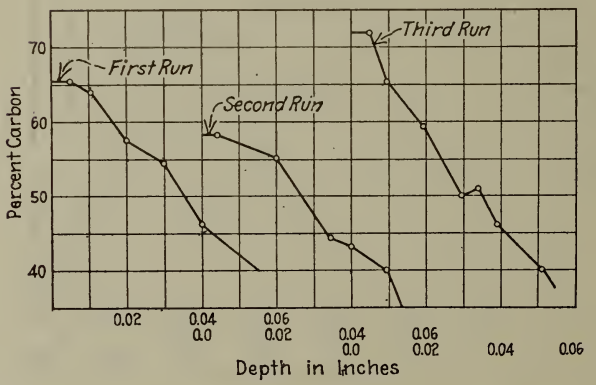

FIG. 7 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 6
PHOSPHORUS CONTENT 0.028%–0.030%; SULPHUR CONTENT 0.034%–0.040%

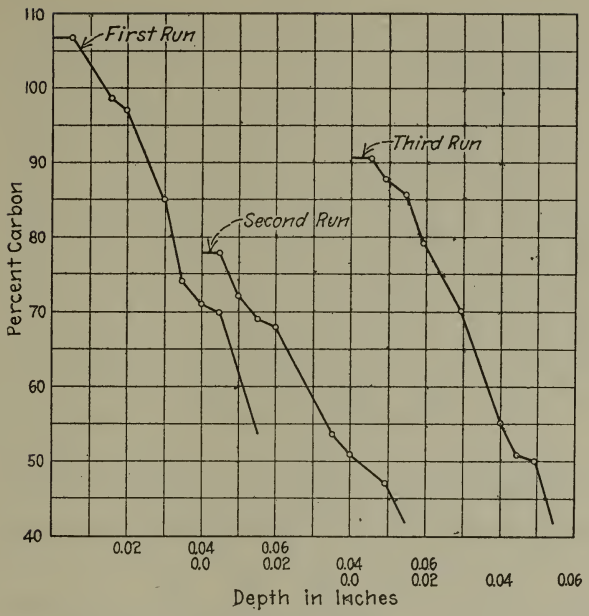

FIG. 8 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 7
PHOSPHORUS CONTENT 0.024%–0.033%; SULPHUR CONTENT 0.040%–0.044%

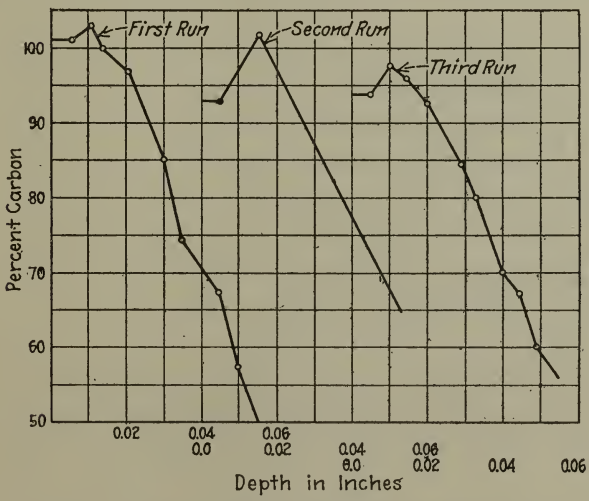

FIG. 9 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 8
PHOSPHORUS CONTENT 0.028%–0.031%; SULPHUR CONTENT 0.042%–0.046%

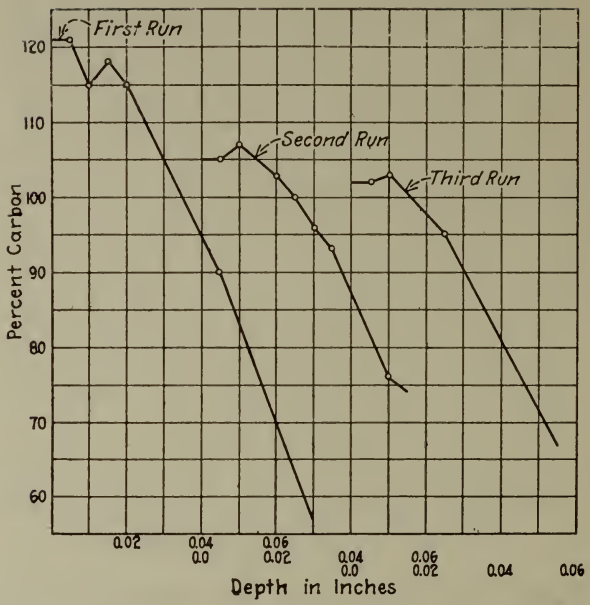

FIG. 10 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 9 (SEE PAR. 44)
PHOSPHORUS CONTENT 0.026%–0.036%; SULPHUR CONTENT 0.050%–0.054%

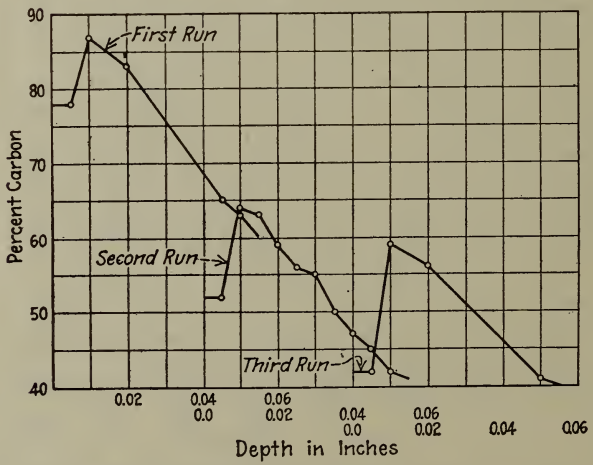

FIG. 11 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 10
PHOSPHORUS CONTENT 0.042%–0.035%; SULPHUR CONTENT 0.044%–0.042%

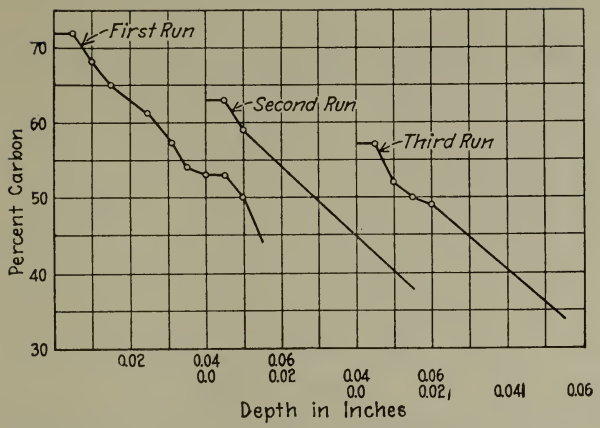

FIG. 12 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 11
PHOSPHORUS CONTENT 0.033%–0.036%; SULPHUR CONTENT 0.042%–0.042%

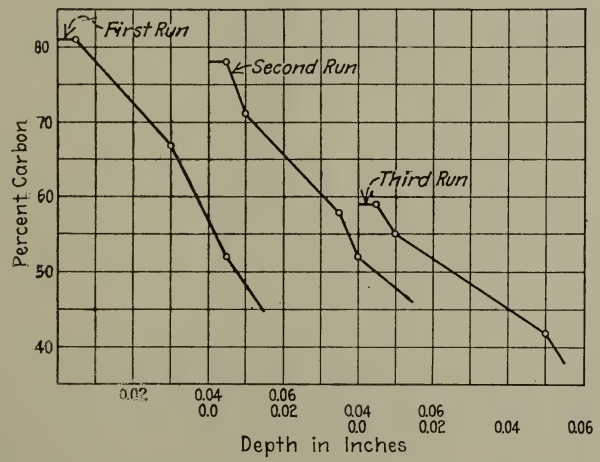

FIG. 13 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 12
PHOSPHORUS CONTENT 0.031%–0.036%; SULPHUR CONTENT 0.044%–0.044%

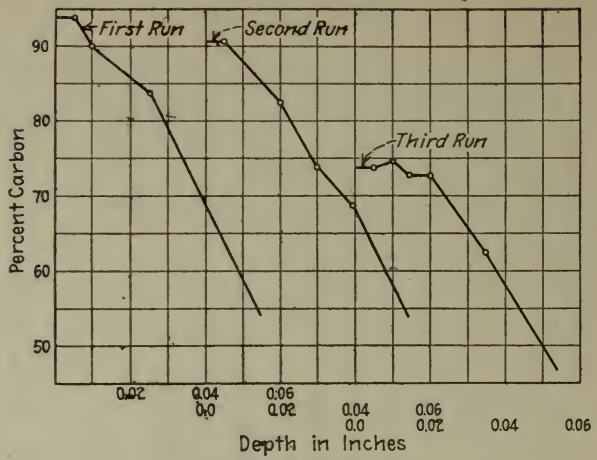

FIG. 14 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 13 (SEE PAR. 44)
PHOSPHORUS CONTENT 0.032%–0.032%; SULPHUR CONTENT 0.044%–0.042%

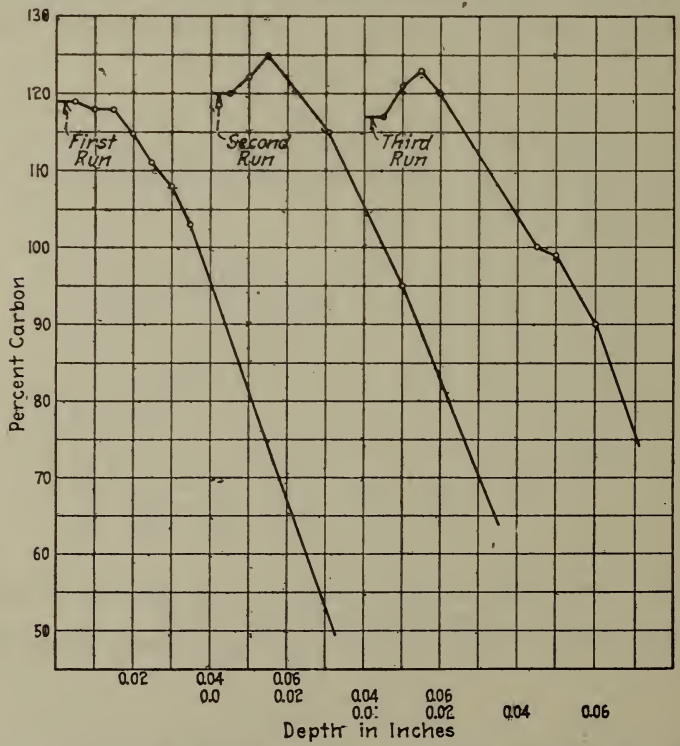

FIG. 15 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 14
PHOSPHORUS CONTENT 0.036%–0.034%; SULPHUR CONTENT 0.036%–0.042%

TABLE 9 PHYSICAL AND PRICE DATA ON CARBONIZING MATERIALS IN RELATION TO CARBON PENETRATION

Carbonizing Material No.	Carbon in Test Bars, Per Cent						Relative C Yielded in Satisfactory Tests per Unit Volume	Price per Cu. Ft.	Cost per Cu. Ft. per Unit
	Depth of 0.005 In.		Depth of 0.025 In.						
	1st	2d	3d	1st	2d	3d			
1	1.02	0.93	0.81	0.87	0.80	....	1.67	$1.84	$1.10
2	1.00	0.91	0.83	0.91	0.76	....	1.67	1.30	0.78
3	0.89	0.85	0.83	....	....	....	....	1.50	....
4	0.96	0.93	0.85	0.88	0.81	....	1.69	1.48	0.87
5	0.96	0.93	0.88	0.85	0.80	....	1.65	1.58	0.96
6	0.65	0.38	0.72	....	....	....	....	0.85	....
7	1.07	0.78	0.91	0.88	0.80	0.73	2.41	1.02	0.42
8	1.01	0.93	0.94	0.85	0.88	0.93	2.56	0.86	0.34
9	1.21	1.05	1.03	0.95	0.90	0.90	2.90	1.64	0.56
10	0.78	0.52	0.43	....	....	....	....	....	....
11	0.72	0.63	0.57	....	....	....	....	1.31	0.85
12	0.81	0.78	0.59	....	....	....	....	....	....
13	0.94	0.91	0.74	0.78	0.78	....	1.56	1.32	5.33
14	1.19	1.20	1.17	1.06	1.13	1.13	3.32	1.77	....

TABLE 10 THERMAL CONDUCTIVITY OF CARBONIZING COMPOUNDS

No.		Price per Ton	Weight per Cu. Ft.	Time to Heat to 1200° F., Min.	Per Cent of Fastest	Per Cent of Slowest
1	New case-carbonizing compound No. 8	$60	28	52	153	34.7
2	Old case-carbonizing compound No. 8	$60	28	34	100	22.7
3	New case-carbonizing compound No. 1	$57	52	145	425	96.5
4	Old case-carbonizing compound No. 1	$57	52	99	291	66.0
5	New case-carbonizing compound No. 14	$90	35	96	282	64.0
6	Old case-carbonizing compound No. 14	$90	35	110	324	73.5
7	New case-carbonizing compound No. 17	$40	51	114	336	76.0
8	Old case-carbonizing compound No. 17	$40	51	76	223	50.5
9	Case-carbonizing compound No. 15	$80	45	112	330	74.5
10	Case-carbonizing compound No. 11	$40	..	143	424	95.5
11	Case-carbonizing compound No. 16	...	..	150	442	100.0

treatment, a compound should be selected which gives a high maximum carbon content.

e When parts are produced which require resistance to shock after heat treatment, a compound should be selected which gives a carbon maximum sufficient for the hardness desired and not in excess of this amount, or brittleness will result.

f The ideal case-hardening compound, as indicated by the experiment, would possess the following characteristics: (1) large volume per ton; (2) small shrinkage per run; (3) high resistivity to change of shape or powdering; (4) cleanliness and freedom from dust; (5) uniform case-carbonizing power at all runs; (6) capability of being used an innumerable number of times.

48 These specifications may seem impossible for any compound. There is now on the market, however, a material which can be used over and over again without loss of carbonizing power, by the addition of new material, which fulfils the conditions mentioned.

EXPERIMENT NO. 6

EXPERIMENT OF THE THERMAL CONDUCTIVITY OF CARBONIZING MATERIALS

49 The object of this experiment was to determine the relative rapidity with which different carbonizing materials would heat up. This factor is of great practical importance in determining the length of time required for the carbonizing operation, inasmuch as previous experiments have shown that, using standard round carbonizing pot 9 in. in diameter and 15 in. high and carbonizing material No. 1, more than two-thirds of the total time needed to attain a depth of 1-32 in. of case under regular carbonizing practice is required in heating up the pot and contents to the carbonizing temperature. It will readily be seen that a saving in time in bringing the contents of the pot up to heat will materially decrease the cost of carbonizing operation.

50 The materials used in conducting this experiment were case-carbonizing materials Nos. 1, 8, 11, 14, 15 and 16. The materials were obtained as samples from the various manufac-

turers. In carrying out the experiment the plan was to heat a lead bath to a constant temperature and to record the time for a thermo-couple in a special pot to come up to some definite temperature. The pot was packed with the different carbonizing mediums. The lead pot was 24 in. long, 15 in. wide, and 7 in. deep, and was filled with lead to a depth so that when the pot containing the carbonizer was immersed in it, the lead stood within ½ in. of the top. The temperature of the lead pot was recorded by a calibrated Le Chatelier thermo-couple. The lead pot was covered by a sheet of steel ¼ in. thick, and containing holes to accommodate the carbonizer pot and thermo-couple tube. The pot containing the carbonizer was made of 6 in. round pipe, ¼ in. wall, and 8½ in. high. A plug ¼ in. thick was welded in the bottom. Angle-iron handles were riveted to opposite sides by means of which the pot was held down in the lead bath. The handles were so located that they held the pot immersed to a depth of 5½ in. in the lead. The temperature of the inside of the pot was recorded by a calibrated Le Chatelier thermo-couple so located that the hot junction was equidistant from the bottom and sides of the pot. Both thermo-couples were connected up to read on the H. H. Franklin Company's single pivot galvanometer through a double throw switch. The temperature of the cold junction of the couples was recorded.

51 In packing the pot with the different carbonizing materials it was filled up level full, after locating the thermo-couple properly, and then tamped in fairly tight. The pot was covered with a thin sheet steel cover containing a hole for the thermo-couple tube and this cover was luted on the clay. Records were taken of the time of putting the packed pot in the lead and every five minutes thereafter of the pot and lead temperatures until the former reached 1200 deg. fahr. The recorded temperatures were corrected for cold junction temperature and the calibration correction and time temperature curves were drawn for all the compounds used.

52 The results of this experiment are summarized in Figs. 16 and 17, which give the numbers of the compounds upon which results are recorded, the time required to heat to 1200 deg. with each compound and the rapidity of rise in temperature shown by the thermo-couple packed in the different carbonizing compounds.

CONCLUSIONS

53 This experiment has served to place the different case-carbonizing materials in a definite order in regard to their rapidity of heating up to case-carbonizing temperatures. The

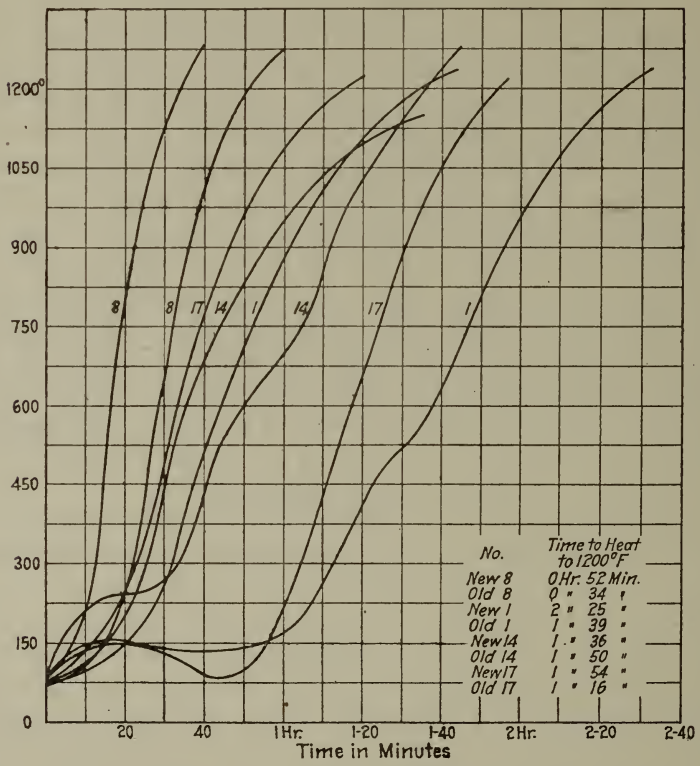

FIG. 16 THERMAL CONDUCTIVITY OF CARBONIZING COMPOUNDS

thermal conductivity of case-carbonizing materials must be carefully considered when low cost for this operation is essential and when large section pots are used. The lag in pots of large section is often sufficient to produce very un-uniform penetration and thus cause greatest warpage and physical variation.

EXPERIMENT NO. 7

EFFICIENCY OF COPPER PLATING IN PREVENTING CARBONIZATION: GENERAL METHOD

54 The object of the experiment was to determine the efficiency of different thicknesses of copper plate in preventing absorption of carbon, and to ascertain the condition under which a plate could be obtained effectually to prevent carbonization.

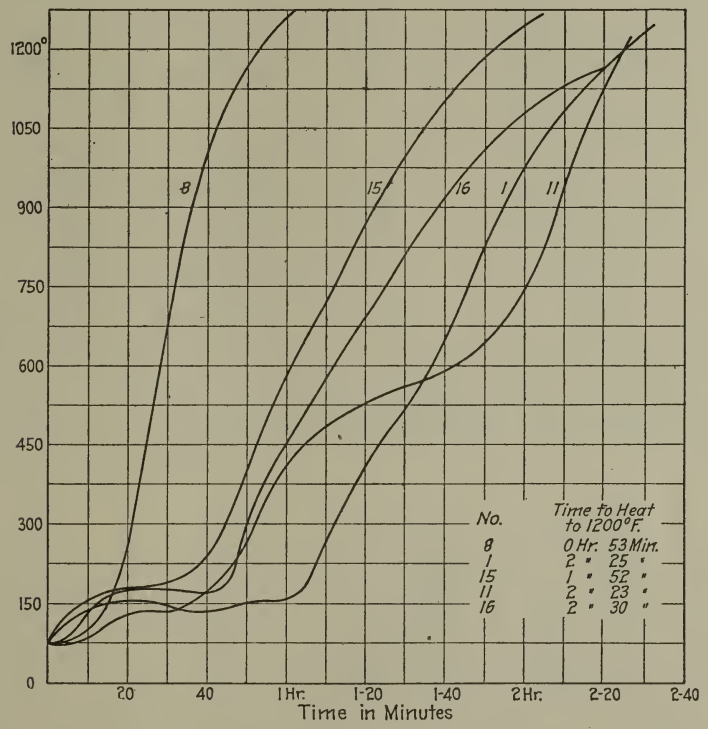

FIG. 17 THERMAL CONDUCTIVITY OF CARBONIZING COMPOUNDS

55 The stock used in the test pieces was steel No. 3 (0.20 per cent carbon basic open-hearth steel). The test pieces were 3½ in. long by 0.450 in. in diameter. They were set up on centers in a lathe and turned to 0.460 in. They were then ground to 0.450 in. and micrometered carefully to 1-10,000 in. All micrometer measurements were made by the same man.

56 The pieces were then copper plated, all conditions of the operation being kept constant except the actual time of plating,

which was carefully noted in each case. After plating the test pieces were micrometered and the thickness of the plate determined. The pieces were then carbonized under standard condition to give a depth of case of 1-32 in. Carbonizing furnace No. 2 was used and the material was case-carbonizing material

TABLE 11 MEASUREMENT[1] OF TEST PIECES; DURATION OF PLATING OPERATION; PER CENT CARBON ABSORBED

No.	Diameter before Copper Plating	Second Length Time in Plating Bath, Sec.	Diameter after Copper Plate	Thickness of Copper Plating
1	0.4496	about 1	0.4497	0.00005
2	0.4500	10	0.4501	0.00005
3	0.4500–0.4498	20	0.4501–0.4499	0.00005
4	0.4502	30	0.4503	0.00005
5	0.4501	60	0.4504	0.00015
6	0.4496	2	0.4500	0.0002
7	0.4500–0.4502	3	0.4507–0.4508	0.0003
8	0.4500	4	0.4509	0.00045
9	0.4499	5	0.4512–0.4509	0.00065–0.0005
10	0.4500	10	0.4525–0.4516	0.00125–0.0008

No.	Diameter after Removal of Copper	Thickness Metal Removed on Radius	Per Cent C Cut No. 1	Per Cent C Cut No. 2
1	0.450 (?)	.....	0.96	0.95
2	0.4504–0.450 (?)	.....	0.58	0.56
3	0.4499–0.4498	0.0002–0.0001	0.40	0.38
4	0.450	0.0003	0.46	0.42
5	0.4498	0.0006	0.20	0.18
6	0.4492–0.4490	0.0008–0.001	0.19	0.17
7	0.4496–0.4498	0.0013–0.001	0.18	0.16
8	0.4493–0.4494	0.0016–0.0015	0.20	0.18
9	0.4494–0.4492	0.0018–0.0016	0.17	0.16
10	0.4495–0.4490	0.003 –0.0026	0.17	0.16

[1] All measurements made by one man and as carefully as possible with a 0.0001 in. micrometer

No. 1, one-half new and one-half used. The pieces were then set up in a lathe and the copper plate removed with emery cloth, after which they were carefully micrometered again.

57 Two cuts were taken on each test piece, each cut being 0.005 in. on the radius, reducing the diameter 0.010 in. The turnings obtained were carefully taken and were very fine. Carbon determinations by direct combustion were made on each cut.

58 Direct electric current for all plating operations was

furnished by a 4-pole direct-current dynamo which is belt driven by an alternating-current constant speed motor. The rated capacity of the dynamo is 360 amperes at 6 volts.

Voltage across terminals.. ..... 4.1
Temperature of bath..... 170 deg. fahr.

Operation No.	Contents of Bath	Purpose
1	Gasolene	To remove grease
2	Sawdust	To dry
3	Warm potassium hydroxide solution	To remove grease and dirt
4	Warm water	To wash
5	Warm sulphuric acid solution	To acid clean
6	Warm water	To wash
7	Cold water	Additional wash
8	Cold potassium cyanide solution	Cleanser
9	Cold water	To wash
10	Electric cleaner, warm sodium hydroxide case-iron anode	Cleanser to give good plating surface
11	Copper plating bath of copper sulphate and potassium cyanide solution warm	Plating bath

59 The copper cover to prevent the penetration entirely must have a thickness of 0.0005 in. Microscopic inspection before and after case carbonizing of copper plated pieces of steel showed that the copper coating was not continuous when the thickness was less than 0.0005 in.

GENERAL CONCLUSIONS

60 This paper may lack interest for the more scientific reader on account of the non-technical nature of the notes, and the want of timely information on the subject of case carbonizing. The information herein presented was obtained with the object of pointing out: (*a*) the quantitative effect of the commercial irregularities which creep into case-carbonized steel parts; (*b*) the precautions which must be observed if uniformity is desired; and (*c*) the physical characteristics in case-carbonizing materials which must be considered if low cost for this operation is to be obtained.

61 The ideal case-carbonizing compound perhaps has not yet been produced; however, compounds which are clean, mechanically strong, uniform in case-carbonizing power, and capable of unlimited re-use are being commercially employed in case-carbonizing operation. This is substantial progress.

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62 The researches are the results of several years' study and the product of the efforts of more than one man. Messrs. Fenner, Kiefer, Baxter and Nead, members of the metallurgical staff of the H. H. Franklin Manufacturing Company, Mr. Bourg, metallurgist for the Brown-Lipe-Chapin Company, and Messrs. H. J. Stagg and C. R. Bulley of the author's staff in the metallurgical department of the Halcomb Steel Company have all contributed their share to the results herein reported and it was only by the coöperation of all that the vast amount of work necessary could be accomplished in so short a time. The author wishes, therefore, to thank these gentlemen and to credit them with their share in the work here reported. The author also wishes to thank Dr. Mathews, manager of the Halcomb Steel Company, for his advice and support. The companies served by the compilers of these data deserve recognition for their progressiveness and their contribution to the general knowledge of the subject in permitting the publication of this report. The subjects of interest connected with the commercial use of steels are so large that it is only by such coöperation that a great deal can be accomplished in a few years.

APPENDIX No. 1

TEMPERATURES AT WHICH TEST PIECES WERE REHEATED OR ANNEALED, EXPERIMENT NO. 1

TABLE 12 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 2

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Quenched from pot at 1500° into water	0.500	900	0.40	0.04
2	Reheat 1400°, quench in water	0.500	775	0.30	0.04
3	Reheat 1500°, quench in water	0.500	740	0.23	0.04
4	Reheat 1600°, quench in water	0.500	850	0.24	0.04
5	Reheat 1700°, quench in water	0.500	800	0.24	0.04
6	Reheat 1800°, quench in water	0.500	815	0.24	0.04
7	Reheat 1600°, quench oil, reheat 1400°, quench water	0.499	750	0.29	0.04
8	Reheat 1600°, quench oil, reheat 1500°, quench oil	0.499	1050	0.50	0.04
9	Reheat 1600°, quench oil, reheat 1500°, quench water	0.500	1000	0.50	0.04
10	Reheat 1500°, quench water, draw 380° for 20 nim.	0.501	820	0.24	0.04
11	Anneal 1400°, reheat 1500°, quench water	0.500	830	0.30	0.04
12	Anneal 1600°, quench oil, reheat 1500°, quench water	0.499	900	0.37	0.04

SUMMARY: 1, Best single heat treatment reheat 1600°, quench water; 2, best double heat treatment reheat 1600°, quench oil, reheat 1500°, quench oil; 3, best double heat treatment increases strength 23 per cent and increases toughness 105 per cent; 4, drawing the temper increases strength 10 per cent and toughness 3 per cent; 5, annealing increases (with single treatment) strength 12 per cent, toughness 25 per cent; 6, annealing decreases (with double treatment) strength 10 per cent, toughness 26 per cent.

TABLE 13 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 3

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1500°, quench in water	0.500	1020	0.61	0.010
2	Reheat 1500°, quench in water	0.500	950	0.44	0.020
3	Reheat 1500°, quench in water	0.500	940	0.29	0.025
4	Reheat 1500°, quench in water	0.500	850	0.31	0.030
5	Reheat 1500°, quench in water	0.500	800	0.30	0.035
6	Reheat 1500°, quench in water	0.500	725	0.24	0.040
7	Quenched from pot at 1500°, in water	0.501	940	0.30	0.040
8	Reheat 1400°, quench in water	0.501	700	0.21	0.040
9	Reheat 1500°, quench in water	0.500	700	0.22	0.040
10	Reheat 1600°, quench in water	0.500	810	0.24	0.040
11	Reheat 1700°, quench in water	0.500	810	0.25	0.040
12	Reheat 1800°, quench in water	0.500	810	0.24	0.040
13	Reheat 1600°, quench oil, reheat 1400°, quench water	0.499	840	0.30	0.040
14	Reheat 1600°, quench oil, reheat 1500°, quench oil	0.498	1000	0.39	0.040
15	Reheat 1600°, quench oil, reheat 1500°, quench water	0.499	970	0.36	0.040
16	Reheat 1500°, quench water, draw 380° for 20 min.	0.500	870	0.26	0.040
17	Anneal 1400°, reheat 1500°, quench water	0.500	845	0.26	0.040
18	Anneal 1600°, reheat 1600°, quench oil, reheat 1500°, quench water	0.500	830	0.26	0.040

SUMMARY: 1, Strength and toughness decrease with increasing depth of case; 2, minimum strength and toughness with 0.040 in. depth of case; 3, best single heat treatment, reheat 1600 deg. fahr., quench water; 4, best double heat treatment, reheat 1600 deg. fahr., quench oil, reheat 1500 deg. fahr., quench oil; 5, double heat treatment increases strength 25 per cent, toughness 56 per cent; 6, drawing temper at 380 deg. fahr. increases strength 21 per cent, toughness 4 per cent; 7, annealing and single treatment increases strength 16.5 per cent, toughness 4 per cent; 8, annealing and double treatment decreases strength 14.5 per cent, toughness 29 per cent.

TABLE 14 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 4

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1500°, quench water	0.500	830	0.48	0.010
2	Reheat 1500°, quench water	0.500	820	0.43	0.015
3	Reheat 1500°, quench water	0.500	740	0.35	0.020
4	Reheat 1500°, quench water	0.500	745	0.34	0.025
5	Reheat 1500°, quench water	0.500	725	0.32	0.030
6	Reheat 1500°, quench water	0.500	700	0.22	0.040
7	Quenched from pot at 1500° in water	0.500	810	0.24	0.040
8	Reheat 1400°, quench water	0.500	670	0.22	0.040
9	Reheat 1500°, quench water	0.500	700	0.22	0.040
10	Reheat 1600°, quench water	0.501	850	0.25	0.040
11	Reheat 1700°, quench water	0.500	830	0.25	0.040
12	Reheat 1800°, quench water	0.500	810	0.24	0.040
13	Reheat 1600°, quench oil, reheat 1400°, quench water	0.500	840	0.30	0.040
14	Reheat 1600°, quench oil, reheat 1500°, quench water	0.500	950	0.45	0.040
15	Reheat 1500°, quench water, draw 380° for 20 min.	0.500	790	0.26	0.040
16	Reheat 1600°, quench oil, reheat 1500°, quench water, draw 380° for 20 min.	0.500	950	0.44	0.040
17	Anneal 1400°, reheat 1500°, quench water	0.500	800	0.26	0.040

SUMMARY: 1, Strength and toughness decrease with increasing depth of case; 2, minimum strength and toughness occur with depth of case of 0.020 in.; 3, best single heat treatment reheat 1600 deg. fahr., quench water; 4, best double heat treatment reheat 1600 deg. fahr., quench oil, reheat 1500 deg. fahr., quench water; 5, double heat treatment increases strength 11.8 per cent, toughness 80 per cent; 6, drawing increases with single treatment strength 12.9 per cent, toughness 18.2 per cent; 7, drawing increases with double treatment strength 0 per cent, toughness 0 per cent; 8, annealing increases with single treatment strength 14.3 per cent, toughness 18.2 per cent.

TABLE 15 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 5

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1550°, quench oil	0.508	1225	2.66	0.014
2	Reheat 1550°, quench oil	0.508	1220	2.30	0.020
3	Reheat 1550°, quench oil	0.509	1215	1.75	0.025
4	Reheat 1550°, quench oil	0.509	1360	1.20	0.030
5	Reheat 1550°, quench oil	0.508	1295	0.70	0.035
6	Reheat 1550°, quench oil	0.509	1255	0.46	0.040
7	Quenched from pot at 1500° into oil	0.505	1400	0.53	0.040
8	Reheat 1310°, quench oil	0.504	850	0.46	0.040
9	Reheat 1400°, quench oil	0.504	1150	0.46	0.040
10	Reheat 1500°, quench oil	0.504	1250	0.46	0.040
11	Reheat 1600°, quench oil	0.504	1250	0.45	0.040
12	Reheat 1700°, quench oil	0.504	1280	0.46	0.040
13	Reheat 1800°, quench oil	0.504	1250	0.43	0.040
14	Reheat 1600°, quench oil, reheat 1400°, quench oil	0.504	1210	0.60	0.040
15	Reheat 1600°, quench oil, reheat 1400°, quench water	0.502	1200	0.42	0.040
16	Reheat 1600°, quench oil, rehcat 1500°, quench oil	0.503	1500	0.51	0.040
17	Reheat 1600°, quench oil, reheat 1500°, quench water	0.504	1240	0.44	0.040
18	Reheat 1500°, quench oil, draw 380° for 20 min	0.504	1560	0.56	0.040
19	Reheat 1600°, quench oil, reheat 1400°, draw 380° for 20 min	0.504	1320	0.72	0.040
20	Anneal 1500°, reheat 1500°, quench oil	0.504	1280	0.43	0.040
21	Reheat 1500°, reheat 1600°, quench oil, reheat 1400°, quench oil	0.505	1200	0.60	0.040

SUMMARY: 1, Strength and toughness decrease with increase depth of case; 2, minimum strength and toughness occur with depth of case 0.040 in.; 3, best single heat treatment reheat 1500°, quench oil; 4, best double heat treatment reheat 1600°, quench oil, reheat 1500°, quench oil; 5, double heat treatment increases strength 20 per cent, toughness 10.8 per cent; 6, drawing increases with single heat treatment strength 24.8 per cent, toughness 10.8 per cent; 7, drawing increases with double heat treatment strength 9.1 per cent, toughness 20 per cent; 8, annealing increases with single heat treatment strength 2.4 per cent, toughness 6.5 per cent; 9, annealing increases with double heat treatment strength 0 per cent, toughness 0 per cent.

TABLE 16 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 6

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1500°, quench oil	0.504	2010	2.40	0.010
2	Reheat 1500°, quench oil	0.504	1675	0.79	0.015
3	Reheat 1500°, quench oil	0.505	1615	0.69	0.020
4	Reheat 1500°, quench oil	0.505	1455	0.55	0.030
5	Reheat 1500°, quench oil	0.505	1390	0.52	0.035
6	Reheat 1500°, quench oil	0.504	1250	0.43	0.040
7	Quenched from pot at 1500° into oil	0.504	1350	0.49	0.040
8	Reheat 1300°, quench oil	0.503	880	0.32	0.040
9	Reheat 1400°, quench oil	0.503	1230	0.42	0.040
10	Reheat 1500°, quench oil	0.504	1315	0.45	0.040
11	Reheat 1600°, quench oil	0.504	1300	0.45	0.040
12	Reheat 1700°, quench oil	0.502	1320	0.45	0.040
13	Reheat 1800°, quench oil	0.502	1300	0.45	0.040
14	Reheat 1600°, quench oil, reheat 1300°, quench oil	0.503	1240	0.75	0.040
15	Reheat 1600°, quench oil, reheat 1400°, quench oil	0.505	1260	0.59	0.040
16	Reheat 1600°, quench oil, reheat 1400°, quench water	0.504	1200	0.37	0.040
17	Reheat 1600°, quench oil, reheat 1500°, quench oil	0.504	1390	0.44	0.040
18	Reheat 1600°, quench oil, reheat 1500°, quench water	0.504	1415	0.46	0.040
19	Reheat 1500°, quench oil, draw 380° for 20 min.	0.504	1610	0.53	0.040
20	Reheat 1600°, quench oil, reheat 1400°, quench oil, draw 380° for 20 min.	0.504	1400	0.65	0.040
21	Anneal 1500°, quench oil	0.504	1370	0.44	0.040
22	Anneal 1600°, quench oil, reheat 1400°, quench oil	0.504	1250	0.60	0.040

SUMMARY: 1, Strength and toughness decreases with increasing depth of case; 2, minimum strength and toughness occurs with depth of case of 0.040 in.; 3, best single heat treatment reheat 1500 deg. fahr., quench in oil; 4, best double heat treatment reheat 1600 deg. fahr., quench in oil, reheat 1500 deg. fahr., quench oil; 5, best double heat treatment increases strength 6.1 per cent and toughness 0 per cent; 6, drawing increases with single treatment strength 23 per cent and toughness 17.8 per cent; 7, drawing increases with double treatment strength 11 per cent and toughness 10.1 per cent; 8, annealing increases with single treatment strength 4.1 per cent and toughness 0 per cent; 9, annealing increases with double treatment strength 0 per cent and toughness 0 per cent.

TABLE 17 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 7

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1500°, quench oil	0.505	1975	2.50	0.010
2	Reheat 1500°, quench oil	0.505	1940	1.25	0.020
3	Reheat 1500°, quench oil	0.505	1760	0.96	0.025
4	Reheat 1500°, quench oil	0.506	1640	0.70	0.030
5	Reheat 1500°, quench oil	0.504	1410	0.53	0.035
6	Reheat 1500°, quench oil	0.505	1275	0.44	0.040
7	Quench from pot at 1500° into oil	0.504	1250	0.42	0.040
8	Reheat 1300°, quench oil	0.501	1000	0.33	0.040
9	Reheat 1400°, quench oil	0.500	1250	0.40	0.040
10	Reheat 1500°, quench oil	0.502	1280	0.44	0.040
11	Reheat 1600°, quench oil	0.502	1280	0.44	0.040
12	Reheat 1700°, quench oil	0.499	1300	0.45	0.040
13	Reheat 1800°, quench oil	0.499	1225	0.42	0.040
14	Reheat 1600°, quench oil, reheat 1300°, quench oil	0.501	1200	0.60	0.040
15	Reheat 1600°, quench oil, reheat 1400°, quench oil	0.500	1320	0.50	0.040
16	Reheat 1600°, quench oil, reheat 1400°, quench water	0.500	1170	0.37	0.040
17	Reheat 1600°, quench oil, reheat 1500°, quench oil	0.500	1380	0.41	0.040
18	Reheat 1600°, quench oil, reheat 1500°, quench water	0.500	1250	0.40	0.040
19	Reheat 1500°, quench oil, draw 380° for 20 min	0.503	1630	0.53	0.040
20	Reheat 1600°, quench oil, reheat 1400°, quench oil, draw 380° for 20 min	0.502	1525	0.58	0.040
21	Anneal 1500°, reheat 1500°, quench oil	0.502	1300	0.40	0.040
22	Anneal 1500°, reheat 1600°, quench oil, reheat 1400°, quench oil	0.502	1300	0.47	0.040

SUMMARY: 1, Toughness and strength decrease with increasing depth of case; 2, minimum strength and toughness occurs with depth of case of 0.040 in.; 3, best single heat treatment reheat 1450 deg. fahr., quench oil; 4, best double heat treatment reheat 1600 deg. fahr., reheat 1400 deg. fahr., quench oil; 5, best double heat treatment increases strength 5.6 per cent and toughness 13.6 per cent; 6, drawing increases with single treatment, strength 27.4 per cent, and toughness 20.5 per cent; 7, drawing increases with double treatment, strength 15.5 per cent, and toughness 16 per cent; 8, annealing increases with single treatment, strength 4 per cent, and toughness 0 per cent; 9, annealing decreases with double treatment, strength 1.5 per cent, and toughness 6 per cent.

TABLE 18 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 8

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1550°, quench oil	0.507	2055	1.14	0.010
2	Reheat 1550°, quench oil	0.508	1625	0.55	0.020
3	Reheat 1550°, quench oil	0.507	1375	0.43	0.040
4	Reheat 1550°, quench oil	0.507	1350	0.42	0.060
5	Reheat 1550°, quench oil	0.506	1350	0.42	0.070
6	Reheat 1550°, quench oil	0.506	1350	0.42	0.095
7	Quenched from pot at 1500° into oil	0.506	1350	0.45	0.040
8	Reheat 1300°, quench oil	0.503	800	0.28	0.040
9	Reheat 1400°, quench oil	0.504	1100	0.34	0.040
10	Reheat 1500°, quench oil	0.503	1250	0.40	0.040
11	Reheat 1600°, quench oil	0.504	1300	0.43	0.040
12	Reheat 1700°, quench oil	0.506	1360	0.44	0.040
13	Reheat 1800°, quench oil	0.504	1250	0.41	0.040
14	Reheat 1550°, quench oil, reheat 1300°, quench oil	0.504	1300	0.48	0.040
15	Reheat 1550°, quench oil, reheat 1400°, quench oil	0.505	1450	0.50	0.040
16	Reheat 1550°, quench oil, reheat 1400°, quench water	0.505	1400	0.39	0.040
17	Reheat 1550°, quench oil, reheat 1500°, quench oil	0.504	1440	0.49	0.040
18	Reheat 1550°, quench oil, reheat 1500°, quench water	0.502	1340	0.37	0.040
19	Reheat 1550°, quench oil, draw 380° for 20 min.	0.503	1750	0.60	0.040
20	Reheat 1500°, quench oil, reheat 1400°, quench oil, draw 380° for 20 min	0.505	1580	0.53	0.040
21	Anneal 1500°, reheat 1500°, quench oil	0.504	1370	0.43	0.040
22	Anneal 1500°, reheat 1550°, quench oil, reheat 1400°, quench oil	0.505	1440	0.46	0.040

SUMMARY: 1, Strength and toughness decrease with increase in depth of case; 2, minimum strength and toughness occur with depth of case 0.040 in.; 3, best single treatment reheat 1500°, quench oil; 4, best double treatment reheat 1550°, quench oil, reheat 1400°, quench oil; 5, best double treatment increases strength 6.9 per cent, toughness 19 per cent; 6, drawing increases with best single treatment, strength 29.6 per cent, toughness 43 per cent; 7, drawing increases with best double treatment 17 per cent, toughness 23.8 per cent; 8, annealing increases with best single treatment 1.5 per cent, toughness 2.4 per cent; 9, annealing decreases with best double treatment, strength 0.7 per cent, toughness 8 per cent.

TABLE 19 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 9

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1500°, quench oil	0.508	1595	0.52	0.010
2	Reheat 1500°, quench oil	0.508	1450	0.46	0.020
3	Reheat 1500°, quench oil	0.509	1450	0.48	0.040
4	Reheat 1500°, quench oil	0.509	1455	0.45	0.060
5	Reheat 1500°, quench oil	0.508	1435	0.47	0.070
6	Reheat 1500°, quench oil	0.509	1400	0.44	0.095
7	Quenched from pot at 1500° into oil	0.504	1700	0.50	0.040
8	Reheat 1300°, quench oil	0.507	1075	0.44	0.040
9	Reheat 1400°, quench oil	0.507	1450	0.44	0.040
10	Reheat 1500°, quench oil	0.507	1450	0.47	0.040
11	Reheat 1600°, quench oil	0.504	1450	0.48	0.040
12	Reheat 1700°, quench oil	0.503	1500	0.50	0.040
13	Reheat 1800°, quench oil	0.503	1500	0.50	0.040
14	Reheat 1550°, quench oil, reheat 1300°, quench oil	0.504	1300	0.47	0.040
15	Reheat 1550°, quench oil, reheat 1400°, quench oil	0.505	1550	0.47	0.040
16	Reheat 1550°, quench oil, reheat 1400°, quench water	0.506	1170	0.32	0.040
17	Reheat 1550°, quench oil, reheat 1500°, quench oil	0.506	1450	0.46	0.040
18	Reheat 1550°, quench oil, reheat 1500°, quench water	0.503	1240	0.34	0.040
19	Reheat 1550°, quench oil, draw 380° for 20 min.	0.506	1950	0.60	0.040
20	Reheat 1560°, quench oil, reheat 1400°, quench oil, draw 380° for 20 min	0.506	1950	0.62	0.040
21	Anneal 1500°, reheat 1500°, quench oil	0.506	1500	0.48	0.040
22	Anneal 1500°, reheat 1550°, quench oil, reheat 1400°, quench oil	0.506	1500	0.48	0.040

SUMMARY: 1, The strength and toughness decreases with increase in depth of case; 2, minimum strength and toughness occur with depth of case of 0.02 in.; 3, best single treatment reheat 1400°, quench oil; 4, best double treatment reheat 1500°, quench oil, reheat 1400°, quench oil; 5, best double treatment increases strength 6.9 per cent and toughness 0 per cent; 6, drawing increases with single treatment, strength 34.4 per cent, toughness 27.6 per cent; 7, drawing increases with double treatment strength 25.8 per cent, toughness 40.5 per cent; 8, annealing increases with single treatment, strength 4.1 per cent, toughness 2.1 per cent; 9, annealing decreases with double treatment, strength 3.3 per cent, toughness 0 per cent.

TABLE 20 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 10

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1700°, quench water	0.505	1700	1.04	0.015
2	Reheat 1700°, quench water	0.507	1630	0.85	0.020
3	Reheat 1700°, quench water	0.505	1600	0.72	0.025
4	Reheat 1700°, quench water	0.508	1550	0.54	0.030
5	Reheat 1700°, quench water	0.507	1410	0.42	0.035
6	Reheat 1700°, quench water	0.510	1385	0.42	0.040
7	Quenched from pot at 1700°	0.506	1490	0.49	0.040
8	Reheat 1400°, quench water	0.506	1150	0.53	0.040
9	Reheat 1500°, quench water	0.506	1200	0.40	0.040
10	Reheat 1600°, quench water	0.506	1400	0.40	0.040
11	Reheat 1700°, quench water	0.506	1400	0.42	0.040
12	Reheat 1800°, quench water	0.506	1390	0.41	0.040
13	Reheat 1700°, quench oil, reheat 1400°, quench water	0.504	1300	1.60	0.040
14	Reheat 1700°, quench oil, reheat 1500°, quench water	0.503	1240	0.38	0.040
15	Reheat 1700°, quench oil, reheat 1600°, quench oil	0.502	1550	0.67	0.040
16	Reheat 1700°, quench oil, reheat 1600°, quench water	0.504	1500	0.44	0.040
17	Reheat 1700°, quench oil, reheat 1700°, quench oil	0.504	1500	0.49	0.040
18	Reheat 1700°, quench oil, reheat 1700°, quench water	0.503	1450	0.48	0.040
19	Reheat 1700°, quench water, draw at 380° for 20 min.	0.504	1600	0.50	0.040
20	Reheat 1700°, quench oil, reheat 1600°, quench water, draw 380° for 20 min	0.504	1750	0.50	0.040
21	Anneal 1550°, reheat 1700°, quench water	0.504	1440	0.43	0.040

SUMMARY: 1, Strength and toughness decrease with increase in depth of case; 2, minimum strength and toughness occur with depth of case of 0.035 per cent; 3, best single heat treatment reheat 1600 deg. fahr. to 1700 deg. fahr., quench water; 4, best double heat treatment reheat 1700 deg. fahr., quench oil, reheat 1600 deg. fahr., quench oil; 5, best double heat treatment increases strength 10.7 per cent, toughness 92.5 per cent; 6, drawing increases with single treatment, strength 14.3 per cent, toughness 25 per cent; 7, drawing increases with double treatment, strength 16.7 per cent, toughness 13.6 per cent; 8, annealing increases with single treatment, strength 2.8 per cent, toughness 7 per cent.

TABLE 21 TRANSVERSE TESTS ON OIL-HARDENED STEELS

STEEL No. 11

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Hardness	Remarks
1	Heated 1700°, quench in oil, drawn 212° for 20 min.	508	2000	0.75	576	.....
2	Heated 1700°, quench in oil, drawn 400° for 20 min.	508	3025	1.55	542	.....
3	Heated 1700°, quench in oil, drawn 600° for 20 min.	508	2375	2.50	508	Did not break

STEEL No. 12

1	Heated 1500°, quenched in oil, drawn 212° for 20 min.	508	1900	0.60	562	.....
2	Heated 1500°, quenched in oil, drawn 400 ° for 20 min.	508	2825	1.50	545	.....
3	Heated 1500°, quenched in oil, drawn 600° for 20 min.	508	2470	2.50	513	Did not break

APPENDIX No 2

TEMPERATURE RECORDS FOR EXPERIMENT NO. 3

TABLE 22 TEMPERATURE RECORD RUN NO. 1, 1300 DEG. FAHR. MATERIAL: BAR 291-2 (0.20% CARBON OPEN HEARTH); BAR 726-2 (0.20% NICKEL): BAR 353-2 (0.20% "A" VANADIUM)

Time	Temperature, Deg. Fahr.				Remarks
	Cold Junction	Furnace	Pot	Pot Corrected	
9.17	90	100	atmos.	....	Furnace started
9.30	97	530	....	....	
9.45	98	880	....	....	
10.00	106	1035	....	....	
10.15	110	1240	490	475	
10.30	106	1400	870	855	
10.45	104	1335	1200	1185	
11.00	108	1320	1310	1295	Begin to count time
11.15	112	1220	1320	1300	
11.30	114	1160	1300	1280	
11.45	118	1225	1260	1240	
12.00	112	1320	1280	1260	
12.15	116	1355	1330	1310	
12.30	...	1360	1330	1310	
12.45	...	1280	1350	1320	
1.00	117	1285	1360	1340	
1.15	118	1255	1330	1310	
1.30	124	1290	1320	1295	
1.45	122	1300	1320	1295	
2.00	117	1255	1335	1315	
2.15	121	1225	1315	1295	
2.30	124	1225	1290	1265	
2.45	124	1345	1310	1280	
3.00	132	1340	1360	1330	
3.15	128	1260	1360	1335	
3.30	120	1200	1330	1305	
3.45	117	1200	1300	1280	
4.00	126	1250	1270	1245	Pot pulled

TABLE 23 TEMPERATURE RECORD RUN NO. 2, 1400 DEG. FAHR.

Time	Temperature, Deg. Fahr.				Remarks
	Cold Junction	Furnace	Pot	Pot Corrected	
8.55	95	400	....	....	Furnace started
9.05	98	500	....	....	
9.15	95	840	....	....	
9.30	100	1040	....	....	
9.45	103	1190	....	....	
10.00	105	1325	715	700	
10.15	105	1445	1150	1135	
10.30	110	1440	1330	1315	
10.45	110	1460	1440	1425	Begin to count time
11.00	112	1400	1460	1440	
11.15	116	1360	1440	1420	
11.30	119	1335	1415	1395	
11.45	114	1385	1410	1390	
12.00	125	1435	1440	1415	
12.15	123	1420	1465	1440	
12.30	120	1400	1460	1435	
12.45	120	1380	1420	1400	
1.00	120	1415	1415	1395	
1.15	125	1345	1400	1375	
1.30	124	1355	1400	1375	
1.45	130	1435	1425	1400	
2.00	130	1440	1450	1425	
2.15	125	1405	1445	1420	
2.30	126	1375	....	....	No. 4 couple off
2.45	...	1320	....	....	
3.00	128	1300	1310	1285	
3.15	132	1410	1325	1290	
3.30	125	1475	1400	1375	
3.45	120	1460	1430	1405	
4.00	125	1420	1425	1400	
4.15	125	1375	1405	1380	Pot pulled

TABLE 24 TEMPERATURE RECORD RUN NO. 3, 1500 DEG. FAHR. MATERIAL: BAR 292-2 (0.20% CARBON OPEN HEARTH); BAR 728-2 (0.20% NICKEL); BAR 354-2 (0.21% "A" VANADIUM)

Time	Temperature, Deg. Fahr.				Remarks
	Cold Junction	Furnace	Pot	Pot Corrected	
8.40	...	150	....	....	Furnace started
9.00	98	650	....	....	
9.15	98	910	....	....	
9.30	101	1090	....	....	
9.45	101	1230	490	475	
10.00	103	1345	910	895	
10.15	106	1440	1190	1175	
10.30	110	1535	1385	1370	
10.45	111	1540	1520	1505	
11.00	112	1540	1570	1550	
11.15	120	1500	1535	1515	
11.30	121	1465	1565	1540	
11.45	118	1460	1525	1505	
12.00	116	1455	1545	1520	
12.15	120	1400	1520	1500	
12.30	120	1360	1280 (?)	1260 (?)	
12.45	120	1420	1400	1380	
1.00	120	1470	1420	1400	
1.15	120	1455	1555	1530	
1.30	126	1370	1530	1505	
1.45	121	1360	1475	1450	
2.00	128	1410	1460	1435	
2.15	132	1480	1525	1495	
2.30	125	1480	1580	1565	
2.45	124	1425	1545	1520	
3.00	128	1380	1510	1485	
3.15	130	1365	1500	1475	
3.30	128	1420	1475	1450	
3.45	122	1500	1520	1496	
4.00	120	1390	1570	1555	
4.15	127	1320	1500	1475	Pot pulled

TABLE 25 TEMPERATURE RECORD RUN NO. 4, 1600 DEG. FAHR. MATERIAL BAR 293-1 (0.20% CARBON OPEN HEARTH); BAR 732-1 (0.20% NICKEL); BAR 355-1 (0.20% "A" VANADIUM)

Time	Temperature, Deg. Fahr.				Remarks
	Cold Junction	Furnace	Pot	Pot Corrected	
10.00	...	....	....	....	Furnace started
10.30	...	1000	....	....	
10.45	...	1210	....	....	
11.00	...	....	....	....	
11.15	80	1460	910	895	
11.30	85	1565	1235	1210	
11.45	86	1655	1465	1445	
12.00	88	1655	1600	1580	
12.15	...	....	....	....	
12.30	...	....	....	....	
12.45	...	....	....	....	
1.00	102	1515	1545	1520	
1.15	99	1565	1565	1540	
1.30	101	1530	1550	1525	
1.45	95	1550	1560	1435	
2.00	84	1625	1600	1580	
2.15	102	1650	1635	1610	
2.30	100	1640	1655	1625	
2.45	90	1580	1655	1625	
3.00	74	1585	1640	1675	
3.15	92	1560	1600	1580	
3.30	90	1555	1500	1475	
3.45	105	1540	1565	1540	
4.00	92	1620	1600	1580	
4.15	...	....	....	....	
4.30	96	1655	1650	1625	
4.45	98	1585	1640	1615	
5.00	98	1530	1595	1575	Pot pulled

TABLE 26 TEMPERATURE RECORD RUN NO. 5, 1700 DEG. FAHR. MATERIAL: BAR 293-2 (0.20% CARBON OPEN HEARTH); BAR 732-2 (0.20% NICKEL); BAR 355-2 (0.20% "A" VANADIUM)

Time	Temperature, Deg. Fahr.				Remarks
	Cold Junction	Furnace	Pot	Pot Corrected	
8.45	...	....	....	....	Furnace started
9.00	...	....	....	....	
9.15	...	1100	....	....	
9.30	...	1300	....	....	
9.45	74	1405	500	980	
10.00	79	1570	980	955	
10.15	76	1700	1375	1330	
10.30	78	1740	1565	1540	
10.45	90	1600	1635	1610	
11.00	88	1590	1610	1590	
11.15	88	1585	1600	1580	
11.30	92	1695	1630	1605	
11.45	102	1800	1740	1725	
12.00	100	1740	1750	1735	
12.15	108	1700	1740	1725	
12.30	...	....	....	....	
12.45	...	....	....	....	
1.00	...	....	....	....	
1.15	91	1650	1665	1650	
1.30	85	1650	1675	1660	
1.45	90	1715	1715	1700	
2.00	91	1685	1720	1705	
2.15	91	1660	1685	1670	
2.30	110	1665	1680	1665	
2.45	90	1665	1685	1670	
3.00	90	1665	1685	1670	
3.15	88	1675	1700	1685	
3.30	104	1675	1680	1665	
3.45	103	1675	1690	1675	
4.00	...	1685	1700	1685	
4.15	100	1695	1705	1690	
4.30	98	1695	1705	1690	
4.45	90	1700	1715	1700	Pot pulled

TABLE 27 TEMPERATURE RECORD RUN NO. 26, 1800 DEG. FAHR. MATERIAL: BAR 284–1 (0.20% CARBON OPEN HEARTH); BAR 469–1 (0.20% NICKEL); BAR 455–1 (0.20% "A" VANADIUM)

Time	Temperature, Deg. Fahr.				Remarks
	Cold Junction	Furnace	Pot	Pot Corrected	
10.00	...	....	....	....	Furnace started
10.15	...	....	....	....	
10.30	...	....	....	....	
10.45	...	1220	....	....	
11.00	...	1380	....	....	
11.15	...	....	....	....	
11.30	88	1620	1215	1200	
11.45	96	1760	1500	1475	
12.00	97	1805	1710	1695	
12.15	109	1810	1780	1780	
12.30	...	....	....	....	
12.45	...	....	....	....	
1.00	108	1865	1870	1860	
1.15	112	1745	1835	1825	
1.30	104	1760	1780	1770	
1.45	106	1805	1810	1800	
2.00	106	1815	1825	1815	
2.15	96	1780	1820	1810	
2.30	...	....	....	....	
2.45	113	1775	1800	1790	
3.00	115	1780	1800	1790	
3.15	105	1780	1805	1795	
3.30	111	1780	1800	1790	
3.45	115	1785	1800	1790	
4.00	102	1785	1805	1795	
4.15	105	1780	1800	1790	
4.30	112	1775	1800	1790	
4.45	101	1770	1795	1785	
5.00	116	1765	1786	1775	Pot pulled

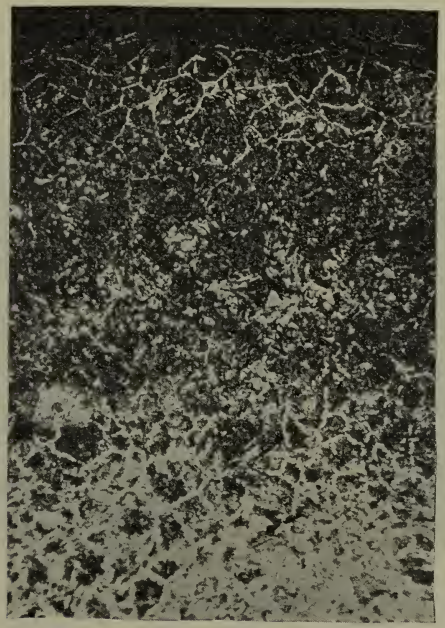

FIG. 18 OPEN HEARTH STEEL, 0.20% CARBON, CASE CARBONIZED AT 1700 DEG. FAHR. SHOWS ENTIRE ZONE AFFECTED BY CASE-CARBONIZING OPERATION; 50 DIAMETERS

In comparison with Figs. 28 and 29 note: (*a*) relatively small carbon penetration; (*b*) mass of cementite needles in outer zone.

FIG. 19 OPEN HEARTH STEEL, 0.20% CARBON, CASE CARBONIZED AT 1300 DEG. FAHR. SHOWS FERRITE BAND PENETRATING CARBONIZED ZONE; 50 DIAMETERS

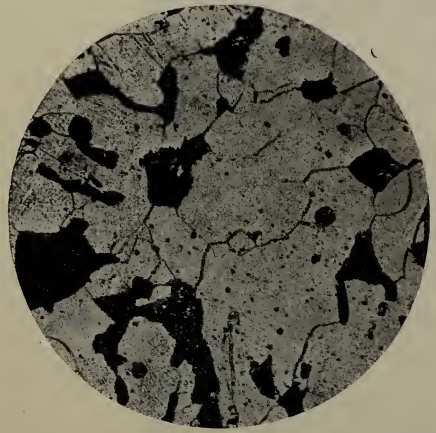

FIG. 20 OPEN HEARTH STEEL, 0.20% CARBON, CASE CARBONIZED AT 1700 DEG. FAHR. STRUCTURE OF METAL AT CORE: 300 DIAMETERS

FIG. 21 OPEN HEARTH STEEL, 0.20% CARBON, CASE CARBONIZED AT 1700 DEG. FAHR. DETAIL OF OUTER EDGE; 300 DIAMETERS

Note thin laminae of cementite which are most likely to cause spawl in service.

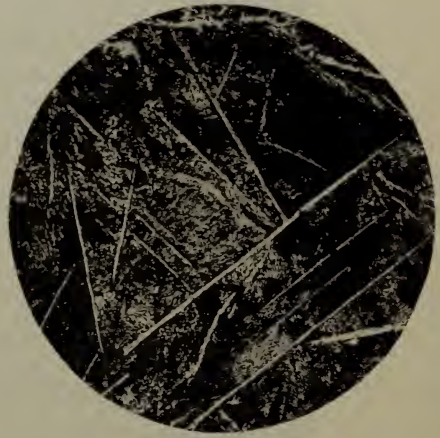

FIG. 22 OPEN HEARTH STEEL, 0.20% CARBON, CASE CARBONIZED AT 1700 DEG. FAHR. VIEW AT POINT SLIGHTLY INSIDE THAT OF FIG. 21; 300 DIAMETERS

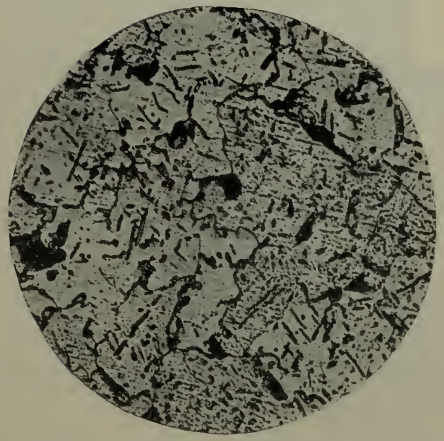

FIG. 23 OPEN HEARTH STEEL, 0.20% CARBON, CASE CARBONIZED AT 1300 DEG. FAHR. NITRIDES AT INNER EDGE OF GRADATION ZONE; 300 DIAMETERS

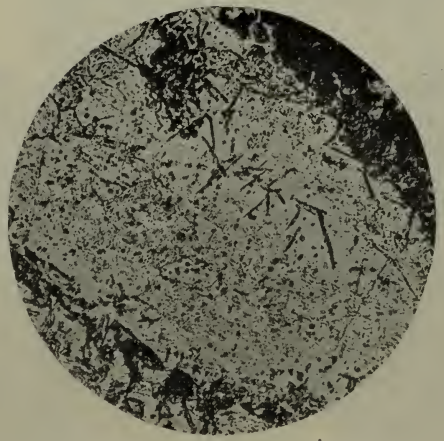

FIG. 24 OPEN HEARTH STEEL, 0.20% CARBON, CASE CARBONIZED AT 1300 DEG. FAHR. NITRIDES IN FERRITE BANDS SHOWN IN FIG. 19; 300 DIAMETERS

Nitrides are black lines seen in white ferrite bands; heavy black mass and black spots are pearlite, details of which could not be shown on account of over printing necessary to bring out nitrides.

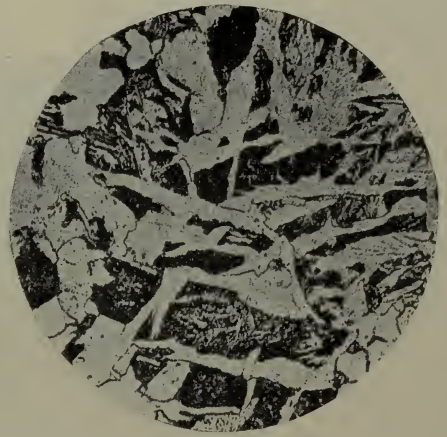

FIG. 25 NICKEL STEEL, 0.20% CARBON, CASE CARBONIZED AT 1700 DEG. FAHR. GENERAL STRUCTURE OF CORE; 300 DIAMETERS

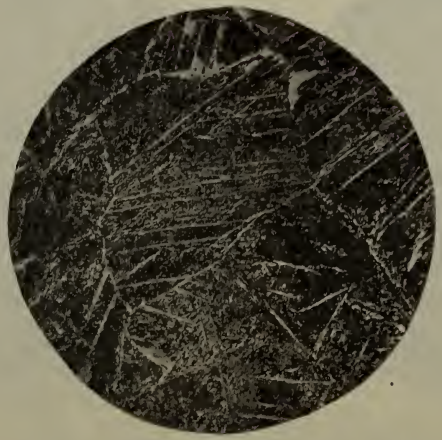

FIG. 26 NICKEL STEEL, 0.20% CARBON, CASE CARBONIZED AT 1700 DEG. FAHR. VIEW OF CEMENTITE NEEDLES TAKEN ABOUT IN MIDDLE OF ZONE OF EXCESS CEMENTITE; 300 DIAMETERS

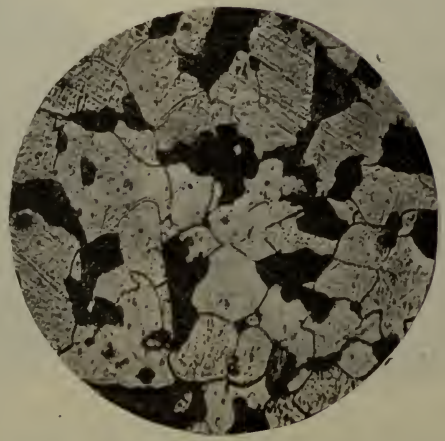

FIG. 27 CHROME VANADIUM STEEL, 0.20% CARBON, CASE CARBONIZED AT 1700 DEG. FAHR. STRUCTURE OF METAL IN CORE; 300 DIAMETERS

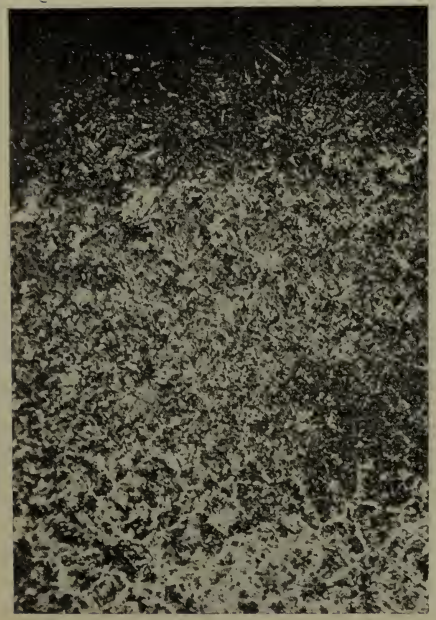

Fig. 28 Nickel Steel, 0.20% Carbon, Case Carbonized at 1700 Deg. Fahr. View shows Deeper Penetration of Carbon in Comparison with Fig. 18; 50 Diameters

Fig. 29 Chrome Vanadium Steel, 0.20% Carbon, Case Carbonized at 1700 Deg. Fahr. Deep Penetration of Carbon and No Needles of Cementite in Outer Zone in Contrast to Figs. 18 and 28; 50 Diameters

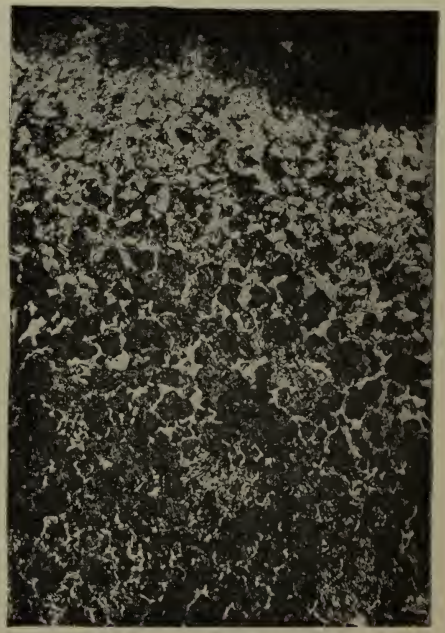

Fig. 30 Chrome Vanadium Steel, 0.20% Carbon, Case Carbonized at 1700 Deg. Fahr. View of Zone of Excess Cementite showing Tendency of Cementite to distribute Itself in Boundaries of Pearlite; 300 Diameters

MICROSCOPIC INVESTIGATION OF OPEN-HEARTH STEELS

TEST PIECE CASE CARBONIZED AT 1300 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 60 SECONDS

63 *Magnification 100 Diameters.* On the outside of this specimen is a zone of pearlite with excess cementite or ferrite in lines too fine for needle test about 0.0075 mm. thick. Inside this is a zone of beautiful lamellar pearlite, about 0.090 mm. thick, of eutectic composition. Further in there is a zone of gradation of carbon content about 0.03375 mm. thick, inside of which is the normal low carbon interior, the pearlite appearing in fine particles surrounded by ferrite. Total penetration of carbon is 0.1350 mm. Just inside the gradation zone and intermingled with it, there is scattered a large number of thin brown needles, varying in size from barely discernible specks to needles about 0.009 mm. long. Mr. K. W. Zimmerschied calls these needles nitrides, since he obtained similar ones by quenching some steel in ammonia. Penetrating the carbonized zone in many places around the circumference are bands of ferrite at various angles from 0 to 90 with the radius containing almost no pearlite, but a thick sprinkling of very minute specks and lines which may be nitrides.

64 *Magnification 400 Diameters.* Under this power the ferrite bands, containing nitrides, appear at intervals all the way around the circumference, more numerous in some parts than in others. These nitrides are more resolved at this power and hence more readily seen, but beyond this increased distinctness, the specimen appears as described above.

TEST PIECE CASE CARBONIZED AT 1400 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 45 SECONDS

65 *Magnification 100 Diameters.* In this specimen the depth of penetration of the carbon seems to be fairly uniform, averaging 0.420 mm. thick. On the outside there is a zone of irregular thickness, containing pearlite with fine lines of excess cementite. The average thickness of this zone is 0.060 mm. Inside the zone of excess cementite comes one of eutectic, about 0.150 mm. thick, followed by a gradation zone 0.225 mm. thick, in which the pearlite exists as large islands surrounded by fine lines of ferrite, growing more numerous toward the interior, until the normal core structure of small islands of pearlite surrounded by ferrite is obtained. The pearlite in the core has gathered into larger islands than in the specimen treated at 1300 deg. fahr. The nitride needles are present inside the eutectic zone, but are not so numerous as in the last specimen. The same is true of the bands of pure ferrite containing nitride needles which penetrate the carbonized zone.

66 *Magnification 400 Diameters.* At this magnification the pearlite throughout the entire carbonized zone is resolved into a beautifully lamellar structure. As before, we have an outer zone of pearlite with excess cementite, showing in places around the circumference small spots of pure cementite on the outer rim. The average thickness of this zone is 0.090 mm. The eutectic zone inside the zone of excess cementite is

approximately 0.120 mm. thick. The gradation zone is 0.1875 mm. thick, giving a total penetration of carbon of 0.3975 mm.

TEST PIECE CASE CARBONIZED AT 1500 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 30 SECONDS

67 *Magnification 100 Diameters.* Under this magnification the effect of carbonizing is noticed in a deeper zone of eutectic, but not in a much greater excess of cementite in the outer portions of the carbonized zone than was observed in the specimen treated at 1400 deg. fahr. On the extreme outer edge of the specimen we see a thin black rim about 0.023 mm. thick, of unresolvable material. Inside this rim we see progressively a zone of pearlite with excess cementite, showing slight crystalline outlines in places, about 0.1275 mm. thick. Next comes the eutectic zone, 0.240 mm. thick, giving a total penetration of carbon of 0.390 mm. This gradation zone varied from large islands of pearlite surrounded by lines of ferrite, to smaller islands of pearlite, surrounded by masses of ferrite. The pearlite on the interior seems to be in larger areas than in the last section surrounded by ferrite. No bands of ferrite, such as supposedly contained nitrides in the last specimens are visible here. Neither are any nitrides visible inside the eutectic zone.

68 *Magnification 400 Diameters.* At this magnification the pearlite in the carbonized zone is resolvable into a beautifully lamellar structure. while that in the core is mostly unresolvable in islands surrounded by ferrite, though some of it appears faintly lamellar. The outer zone of pearlite with excess cementite was 0.150 mm. thick. Inside this was a zone of eutectic 0.150 mm. thick, and then a gradation zone 0.420 mm. thick, giving a total carbon penetration of 0.720 mm. Along the inner part of the gradation zone, we come across a needle or two as we follow the zone around, which might be a nitride, but beyond these few solitary needles no traces, such as the bands cutting into the case, are visible.

TEST PIECE CASE CARBONIZED AT 1600 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 20 SECONDS

69 *Magnification 100 Diameters.* The carbonizing at this temperature appears to have increased the depth of the carbonized zone, but not to have formed a zone of much greater percentage excess cementite than was obtained at the last temperature. The thickness of the outer zone of pearlite and excess cementite varies around the circumference, but averages 0.0975 mm. Inside this zone we find a zone of eutectic averaging 0.345 mm. thick, and a zone of gradation averaging 0.300 mm., giving a total penetration of carbon of 0.7498 mm.

70 *Magnification 400 Diameters.* The pearlite is distinctly lamellar at this magnification, both in the carbonized zone, and in smaller laminations in the interior. The zone of excess cementite varies in thickness, but averages 0.135 mm. Inside this zone comes one of eutectic approximately 0.210 mm. thick, very irregular in depth, and then a gradation zone averaging 0.450 mm., giving a total penetration of carbon of 0.795 mm.

TEST PIECE CASE CARBONIZED AT 1700 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 20 SECONDS

71 *Magnification 100 Diameters.* In this specimen we get both a deeper penetration of carbon than in the last specimen, and a thicker zone of excess cementite. The zone of pearlite with excess cementite was very irregular in depth, but averaged 0.390 mm. In this zone we see a few long cementite needles within the pearlite of varying length and direction, but all of them in straight lines. Inside of this zone comes one of eutectic approximately 0.300 mm. thick, then a gradation zone of about 1.0725 mm., giving a total carbon penetration of 1.7625 mm.

72 *Magnification 400 Diameters.* At this magnification the pearlite appears in large laminations in the carbonized zone and in smaller laminations in the interior. The outer zone, containing pearlite with excess cementite, is approximately 0.225 mm. thick; inside of which comes the eutectic zone, approximately 0.338 mm. thick, and following this a gradation zone 0.60 mm. thick. The outer zone of pearlite with excess cementite contains a large number of cementite needles scattered through it. These needles are all straight lines but point in every direction, the largest of them being about 0.135 mm. long, the widest not over 0.00150 mm. wide. Both the excess cementite and the excess ferrite in their respective zones follow more or less the irregular crystal boundaries.

TEST PIECE CASE CARBONIZED AT 1800 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 15 SECONDS

73 *Magnification 100 Diameters.* In this specimen the excess cementite in the outer zone does not appear so much in the form of crystalline boundaries as it did in the preceding specimen, but more as irregular areas along the outer portion of the zone. The average thickness of this zone is about 0.1950 mm. Inside this zone comes the zone of eutectic, averaging approximately 0.585 mm., followed by a gradation zone averaging 0.975 mm., giving a total penetration of carbon of 1.755 mm. As before, the ferrite in the gradation zone appears in the form of outlines of crystals.

74 *Magnification 400 Diameters.* In this specimen the pearlite is beautifully lamellar in the carbonized zone and lamellar to a slight extent in the interior. On account of the great irregularity in the depth of penetration in the excess cementite in the outer zone, it is difficult at this magnification to obtain an average depth. The quantity of cementite needles within the pearlite grains has increased. The following figures are approximate: For the zone of excess cementite the depth is 0.293 mm. The depth of the eutectic zone is 0.450 mm. and that of the gradation zone 0.90 mm., giving a total penetration of carbon of 1.64 mm.

MICROSCOPIC INVESTIGATION OF CASE-CARBONIZED NICKEL STEELS

TEST PIECE CASE CARBONIZED AT 1300 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 40 SECONDS

75 *Magnification 100 Diameters.* At this magnification this specimen shows only a slight carbonization on the outside, apparently not even reaching the eutectic. In places around the circumference bands of darker material, looking like eutectic, extend quite a little distance into the section, very similar in appearance to the bands containing nitrides noticed in the open-hearth series. These bands occur only in two or three places. The interior of the section consists of finely divided pearlite with excess ferrite. Since the effect of carbonization is slight, it is difficult to tell where the carbonized zone ends and the normal interior structure begins, but an approximate figure of 0.045 mm. thickness gives a general idea of the depth of penetration of the carbon.

76 *Magnification 400 Diameters.* On the flat outer rim of this section appears to be a little pure ferrite, but this is a very thin rim, so that a needle test does not distinguish it from cementite. The pearlite in the carbonized zone is not resolvable at this power. The average depth of the carbonized zone is 0.180 mm. and no eutectic is present.

TEST PIECE CASE CARBONIZED AT 1400 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 40 SECONDS

77 *Magnification 100 Diameters.* The outer zone at this magnification appears to be of eutectic composition and an average thickness of 0.195 mm., inside of which comes a gradation zone approximately 0.390 mm. thick. The pearlite in the interior has gathered together slightly more than in the last specimen, and as before, is surrounded by ferrite. In this specimen we see ferrite bands at different places around the circumference penetrating the outer zone, which look almost the same as those found in the open-hearth series, especially in the specimen treated at this temperature, which contains nitrides.

78 *Magnification 400 Diameters.* The pearlite in the carbonized zone is faintly resolvable into the lamellar condition at this magnification. The highest carbon appears to be eutectic in a zone at the outside of the piece approximately 0.203 mm. thick. In one or two places there appear to be a slight excess of ferrite on the very outside, but this may be due to the rounding of the edges. The bands already mentioned, which penetrated the carbonized zone, are ferrite containing either nitrides or pearlite. At this power no needles which definitely resemble the nitrides are visible.

TEST PIECE CASE CARBONIZED AT 1500 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 20 SECONDS

79 *Magnification 100 Diameters.* In this specimen we get an irregular carbonized zone apparently not going beyond the eutectic in carbon content on the outside. The zone of eutectic, however, is much thicker than in the last specimen. The bands, which penetrated the carbonized zone

in the last specimen are here present in about the same number. The approximate thickness of the eutectic zone is 0.338 mm. The gradation zone is 0.585 mm., giving a total carbon penetration of 0.923 mm.

80 *Magnification 400 Diameters.* At this power we see an outer zone of eutectic approximately 0.278 mm. and a gradation zone of 0.510 mm. The total penetration of carbon is approximately 0.788 mm. The pearlite on the interior is just discernible as lamellar at this magnification and that in the carbonized zone is a little more easily recognized and also lamellar. The bands penetrating the eutectic on the outside appear to consist of pure ferrite with dark dots and smaller irregular lines scattered throughout.

TEST PIECE CASE CARBONIZED AT 1600 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 15 SECONDS

81 *Magnification 100 Diameters.* At this magnification there is little, if any, excess cementite in the outer zone of carbonization. There is, however, a thick zone of eutectic and inside this a zone of decreasing carbon. Also, in one or two places around the circumference we see the penetration bands which have been discussed in the last two or three specimens. The average thickness of the eutectic is 0.390 mm. That of the gradation zone is 0.683 mm. The total penetration of carbon is 1.073 mm.

82 *Magnification 400 Diameters.* At this magnification there appears to be a little excess cementite on the outer edge of the eutectic zone in places. This cementite appears in very fine lines like outlines of ferrite crystals. Since the cementite penetrates to varying depths and is present only in a small amount, we will consider the zone of eutectic as running out to the edge of the specimen and give for an average thickness of eutectic 0.338 mm., inside of which comes a gradation zone approximately 0.563 mm. The total penetration of carbon is 0.901 mm. The pearlite in the carbonized zone is resolvable in places into laminations. The maximum penetration of the excess cementite is about 0.180 mm.

TEST PIECE CASE CARBONIZED AT 1700 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 15 SECONDS

83 *Magnification 100 Diameters.* In this specimen the outer zone, containing pearlite with excess cementite, shows the needles of cementite scattered through it pretty freely, and as lines in the form of crystalline division. The whole outer zone, with the exception of the cementite needles and a few small white spots of cementite, appears to be pearlite. The thickness of this zone is 0.760 mm. Inside of this comes a gradation zone approximately 0.975 mm. The total penetration of carbon is 1.755 mm. No bands such as were seen in the last specimens and which penetrated the carbonized zone appear here, except in one spot where there is a slight penetration, the material looking like cementite.

84 *Magnification 400 Diameters.* At this power the zone of excess cementite containing needles of cementite is readily distinguished, and though varying in depth is approximately 0.473 mm. thick, inside of which

comes a zone of eutectic about 0.225 mm. thick. Inside of this is a gradation zone 0.675 mm. thick. The total penetration of carbon is 1.373 mm. The two bands of cementite already mentioned look like those described under the last specimen and may contain either this peculiar form of pearlite or nitrides.

TEST PIECE CASE CARBONIZED AT 1800 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 15 SECONDS

85 *Magnification 100 Diameters.* This specimen shows a greater penetration of the excess cementite than the last one, but the cementite appears in the form of irregular globules rather than as crystalline boundaries, in which form it appeared in the open-hearth series. The average depth of the zone of excess cementite is 0.293 mm. inside of which is the eutectic zone approximately 0.488 mm. thick, and inside of which a zone of decreasing carbon content approximately 1.28 mm. The total penetration of carbon is 2.061 mm. The bands penetrating the crystalline zone which appear in the last specimens are not present.

86 *Magnification 400 Diameters.* At this magnification the excess cementite in the outer zone is easily seen and appears principally as globules and finer lines, though only a few needles are present. The thickness of this zone is approximately 0.450 mm., inside of which comes a zone of eutectic approximately 0.428 mm. followed by a zone of decreasing carbon 0.15 mm., giving a total carbon penetration of 2.002 mm. The pearlite in the carbonized zone is only slightly resolvable at this magnification into lamellar pearlite. That on the interior is not resolvable at this power and may be troostitic.

MICROSCOPIC INVESTIGATION OF CHROME VANADIUM STEEL

TEST PIECE CASE CARBONIZED AT 1300 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 45 SECONDS

87 *Magnification 100 Diameters.* On the outside of this specimen is a black etching zone, unresolvable at this power, but probably eutectic pearlite, about 0.03 mm. thick, inside of which is a gradation zone 0.075 mm. thick. Inside these two zones is a black etching zone approximately 0.0675 mm. thick, which appears to be pearlite gathered together more or less in areas surrounded by lines of ferrite. The thicknesses of these individual zones varied but the carbon penetration was pretty uniform and averaged 0.1725 mm.

88 *Magnification 400 Diameters.* On the outside there is a zone of pearlite containing small white globules that are probably cementite, though they are too small to apply the needle test, to distinguish them from ferrite. In places there is a very thin outer rim of cementite which confirms the above statement regarding the white globules. Inside this is a zone of very dark etching, faintly resolvable lamellar eutectic pearlite, approximately 0.03 mm. thick, inside of which is a gradation zone about 0.01575 mm. thick. This zone is difficult to measure since the carbon content grades off very slowly.

TEST PIECE CASE CARBONIZED AT 1400 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 30 SECONDS

89 *Magnification 100 Diameters.* On the outside there is a thick zone of eutectic showing possibly a slight excess cementite in fine lines on the outer edge in some places, the average thickness of the zone being about 0.15 mm. Inside of this zone is a gradation zone approximately 0.2475 mm. thick. As in the last specimen the decrease in carbon content is very gradual, making this zone difficult to measure. Total penetration of carbon is 0.03975 mm.

90 *Magnification 400 Diameters.* Here we see on the outside a thin rim about 0.0090 mm. thick of pearlite in excess cementite, inside of which comes a eutectic zone about 0.135 mm. thick, and then a gradation zone about 0.300 mm. thick, giving a carbon penetration of 0.444 mm.

TEST PIECE CASE CARBONIZED AT 1500 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 20 SECONDS

91 *Magnification 100 Diameters.* This specimen shows on the outside a definite zone of pearlite with excess cementite about 0.0525 mm. thick, inside of which is a eutectic zone about 0.4875 mm. and then a gradation zone of 0.195 mm. thick. Total penetration of carbon is 0.735 mm.

92 *Magnification 400 Diameters.* At this magnification there is a zone of pearlite with large excess cementite. This zone is about 0.06 mm. thick. inside it is a zone of pearlite, with just a little excess cementite, in small polygonal outlines, the average thickness of the zone being about 0.225 mm. thick, and then a eutectic zone about 0.180 mm. thick, followed by a gradation zone approximately 0.3375 mm. thick. The pearlite is not resolvable either in the carbonized zone or in the core. Total penetration of carbon is 0.8025 mm.

TEST PIECE CASE CARBONIZED AT 1600 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 150 SECONDS

93 *Magnification 100 Diameters.* Here we see a zone of pearlite with a large excess of cementite about 0.0375 mm., inside of which is a zone of eutectic about 0.51 mm. thick, and then a gradation zone 0.248 mm. thick. Total penetration of carbon is 0.795 mm.

94 *Magnification 400 Diameters.* With this magnification we see an outer zone about 0.045 mm. thick, consisting of cementite with pearlite in small areas throughout it. Then we come to a zone of pearlite containing excess cementite in fine polygonal outlines, this zone being about 0.1125 mm. Inside this zone is a eutectic zone approximately 0.2475 mm. thick, followed by a gradation zone approximately 0.2775 mm. thick, giving a total penetration of carbon of about 0.6825 mm. The pearlite in the carbonized zone is mostly unresolvable, though some of it shows very fine laminations. That on the interior is resolvable into fine lamellar pearlite.

TEST PIECE CASE CARBONIZED AT 1700 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 15 SECONDS

95 *Magnification 100 Diameters.* On the outside of this specimen we see a sharply defined zone of pearlite with excess cementite approximately

0.067 mm. thick. Next comes a zone of eutectic about 0.8755 mm. thick, followed by a gradation zone about 0.4875 mm. thick. Total penetration of carbon is 1.430 mm.

96 *Magnification 400 Diameters.* Here, as in the last specimen, we get a sharply defined zone of cementite with small areas of pearlite scattered through it, apparently a little more pearlite than half and half, this zone being about 0.105 mm. thick. Inside this is a zone about 0.360 mm. thick, which contains pearlite surrounded by thin lines of cementite in polygonal outlines. Then comes the eutectic zone about 0.405 mm. thick, followed by a zone of gradation approximately 0.45 mm. thick. The carbon content in the latter changes very slowly so that the zone is difficult to measure. Total penetration of carbon is 1.320 mm.

TEST PIECE CASE CARBONIZED AT 1800 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 15 SECONDS

97 *Magnification 100 Diameters.* This specimen shows a zone on the outside of pearlite surrounded by fine lines of cementite, this zone being about 0.15 mm. thick. There comes a zone of eutectic approximately 0.975 mm. thick, followed by a gradation zone approximately 0.795 mm. thick, giving a total carbon penetration of 1.92 mm. In various places throughout the carbonized zone appear irregularly shaped areas not over 0.675 mm. approximately, or their longest dimension, which are unresolvable at this power.

98 *Magnification 400 Diameters.* At this magnification, we see a wide zone, about 0.3375 mm. thick, containing pearlite with excess cementite in fine lines showing polygonal figures, like boundaries between ferrite crystals. Inside this comes a zone of eutectic 0.5625 thick, followed by a gradation zone 0.675 mm. thick, giving a total penetration of carbon of 1.575 mm. The white spots above noticed were shown to be cementite by scratching with a needle. In some places there are little white spots containing fine brown lines possibly looking like lamellar pearlite, in which the ferrite layer is very thin, much thinner than the cementite, in some cases only one-seventh to one-tenth as wide. This structure has been noticed in nearly all the areas containing excess cementite.

APPENDIX No. 3

EFFICIENCY OF CASE-CARBONIZING MATERIALS

CASE-CARBONIZING MATERIAL NO. 1

Etched in 2.5 Per Cent Picric Acid for 45 Minutes

99 *Run No. 1, Bar No. 1.* On the outside of this specimen is a uniform zone 0.455 mm. thick of pearlite containing lines of excess cementite; inside is a zone of eutectic pearlite 0.455 mm. thick, followed by a gradation zone 0.650 mm. thick, inside of which comes the normal structure of about 0.20 per cent carbon steel consisting of ferrite containing small islands of pearlite. The pearlite throughout the section is lamellar. Total penetration of carbon is 1.560 mm.

100 *Run No. 2, Bar No. 7.* This specimen shows the outer zone of eutectic pearlite 0.715 mm. thick, inside of which comes a gradation zone 1.001 mm. thick, followed by the normal interior of ferrite containing islands of pearlite. The pearlite throughout the section is lamellar. Total penetration of carbon is 1.716 mm.

Etched in 2.5 Per Cent Picric Acid for 30 Seconds

101 *Run No. 3, Bar No. 13.* This specimen shows an outer zone of eutectic 0.358 mm. thick, inside of which is a gradation zone 1.138 mm. thick, surrounding the normal low carbon interior structure. Total penetration of carbon is 1.496 mm.

Etched in 2.5 Per Cent Picric Acid for 60 Seconds

102 *Run No. 1, Bar No. 2.* This specimen shows an outer zone of lamellar pearlite containing lines of excess cementite, the zone being 0.450 mm. thick. Inside this zone comes a zone of eutectic 0.650 mm. thick, followed by a gradation zone 0.748 mm. thick, giving a total penetration of carbon of 1.848 mm. Inside these zones is the normal low carbon structure.

103 *Run No. 2, Bar No. 8.* On the outside of this specimen is a zone of eutectic 0.650 mm. thick, inside of which comes a gradation zone 1.073 mm. giving a total penetration of carbon of 1.723 mm. The outer zone contains numerous small white areas, probably of ferrite, but too small to scratch with the needle.

Etched in 2.5 Per Cent Picric Acid for 70 Seconds

104 *Run No. 3, Bar No. 14.* In this specimen the ferrite penetrates to the very edge in many places around the circumference and the effect of the carbonizing has been simply to raise the carbon content of the outer portion, forming a zone of decreasing carbon about 0.975 mm. thick.

CASE-CARBONIZING MATERIAL NO. 3

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 45 SECONDS

105 *Run No. 1, Bar No. 3.* Outer zone of eutectic is 0.520 mm. thick; inside this is a zone of gradation 1.235 mm. thick, giving a total penetration of carbon of 1.755 mm. The outer zone contains small white spots looking like ferrite, but too small to scratch.

106 *Run No. 2, Bar No. 9.* On the outside a rather irregular zone of eutectic approximately 0.520 mm. thick, inside of which is a gradation zone 1.300 mm. giving a total penetration of carbon of 1.820 mm. thick. The outer zone of eutectic contains here and there small islands probably of ferrite, too small to scratch with the needle.

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 60 SECONDS

107 *Run No. 3, Bar No. 15.* Outer zone of eutectic in places, but penetrated quite often along the circumference by ferrite bands so that it is questionable whether even the very outside has reached the carbon content of the eutectic. From the outside the carbon content decreases to that of the normal low carbon interior, this gradation zone being 0.325 mm. thick, which is the total penetration of carbon.

CASE-CARBONIZING MATERIAL NO. 1

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 60 SECONDS

108 *Run No. 1, Bar No. 4.* On the outside of this specimen is a zone of eutectic 0.975 mm. thick, inside of which is a gradation zone 1.300 mm. thick, giving a total penetration of carbon of 2.275 mm. In places along the outer zone there are small islands of white material, either cementite or ferrite too small to scratch with the needle and containing very fine lines of pearlite in lamellation.

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 50 SECONDS

109 *Run No. 2, Bar No. 10.* On the outside is a zone of eutectic approximately 0.878 mm. thick, inside of which comes a zone of gradation 0.975 mm. thick, giving a total penetration of 1.853 mm. As in the other specimens there are small spots of white material scattered through the outer zone.

110 *Run No. 3, Bar No. 16.* On the outside is a zone of eutectic approximately 0.455 mm. thick, quite irregular and in places having ferrite penetrating it to the edge of the specimen. Inside of this zone comes one of decreasing carbon 0.975 mm. thick, giving a total penetration of carbon of 1.43 mm.

CASE-CARBONIZING MATERIAL NO. 5

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 60 SECONDS

111 *Run No. 1, Bar No. 5.* On the outside of this specimen is a zone of lamellar pearlite containing lines of excess cementite in polygonal outline as usual, this zone being about 0.780 mm. thick; inside comes the eutectic zone 0.520 mm. thick, followed by a zone of gradation 0.975 mm. thick, giving a total penetration of carbon of 2.275 mm.

112 *Run No. 2, Bar No. 11.* On the outside is a eutectic zone approximately 0.780 mm. thick but quite irregular in depth. Then comes a gradation zone 0.975 mm. thick, giving a total penetration of carbon of 1.755 mm. As usual in the outside zone are small white spots of ferrite or cementite too small to scratch with the needle.

113 *Run No. 3, Bar No. 17.* In this specimen is an outer zone of eutectic 0.225 mm. thick, containing several white spots of ferrite, inside of which comes a gradation zone of 1.300 mm. thick, giving a total penetration of carbon of 1.525 mm.

CASE-CARBONIZING MATERIAL NO. 6

Etched in 2.5 Per Cent Picric Acid for 55 Seconds

114 *Run No. 1, Bar No. 6.* This carbonizing compound did not even bring the outer skin of the metal up to eutectic but carbonized the outside slightly to a depth of 1.235 mm., which is the total penetration of carbon.

Etched in 2.5 Per Cent Picric Acid for 60 Seconds

115 *Run No. 2, Bar No. 12.* This specimen shows an outer zone of increased carbon content not as high as eutectic, the zone being 0.975 mm. thick, which is the total penetration of carbon.

116 *Run No. 3, Bar No. 18.* On the outside of this specimen is a zone of increased carbon content not as high as eutectic, 0.040 mm. thick.

CASE-CARBONIZING MATERIAL NO. 7

Etched in 2.5 Per Cent Picric Acid for 55 Seconds

117 *Run No. 1, Bar No. 19.* On the outside of this specimen is a zone of lamellar pearlite containing lines of excess cementite more or less in polygons, the depth of the zone being 0.325 mm. thick, inside this comes a eutectic zone 0.520 mm. thick, and inside this a gradation zone 0.975 mm. thick, giving a total penetration of carbon of 1.920 mm.

Etched in 2.5 Per Cent Picric Acid for 50 Seconds

118 *Run No. 2, Bar No. 25.* On the outside is a zone of eutectic approximately 2.60 mm. thick, containing small spots of white material probably ferrite in one or two places; inside comes a gradation zone 1.300 mm. thick, giving a total penetration of carbon of 1.560 mm.

Etched in 2.5 Per Cent Picric Acid for 60 Seconds

119 *Run No. 3, Bar No. 31.* On the outside is a zone of eutectic approximately 0.715 mm. thick, containing a little excess ferrite in spots around it, in some places the ferrite extending almost to the edge. Inside of this comes a gradation zone 1.170 mm. thick, giving a total penetration of carbon of 1.995 mm.

CASE-CARBONIZING MATERIAL NO. 8

Etched in 2.5 Per Cent Picric Acid for 50 Seconds

120 *Run No. 1, Bar No. 20.* This specimen shows an outer zone of lamellar pearlite containing polygonal lines of excess cementite, the zone

being approximately 0.455 mm. thick; inside comes a eutectic zone 0.520 mm. thick and a gradation zone 0.975 mm. thick, giving a total penetration of carbon of 1.950 mm.

121 *Run No. 2, Bar No. 26.* On the outside is a zone of lamellar pearlite containing fine lines of excess cementite following the boundaries of the ferrite crystals as usual, the thickness of the zone averaging 0.325 mm.; inside is a zone of eutectic 0.715 mm. thick, followed by a gradation zone 0.910 mm. thick, giving a total penetration of carbon of 1.950 mm.

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 55 SECONDS

122 *Run No. 3, Bar No. 32.* On the outside is a zone of lamellar pearlite containing a little excess cementite in very fine lines following the boundaries of the ferrite crystals. The thickness of this zone is 0.195 mm. Inside of this comes a eutectic zone 0.650 mm. thick, followed by a gradation zone 0.650 mm. thick, giving a total penetration of carbon of 1.495 mm.

CASE-CARBONIZING MATERIAL NO. 9

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 45 SECONDS

123 *Run No. 1, Bar No. 21.* On the outside of this specimen is a zone 1.235 mm. thick of lamellar pearlite containing lines of excess cementite in rather thick lines around the ferrite crystals and showing small cementite needles in places, which appear to branch off from the main crystalline boundaries. Inside comes a eutectic zone 0.423 mm. thick, followed by a gradation zone 0.780 mm. thick, giving a total penetration of carbon of 2.438 mm.

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 55 SECONDS

124 *Run No. 2, Bar No. 27.* On the outside is a zone 1.075 mm. thick of lamellar pearlite containing lots of excess cementite in polygonal outlines, but no cementite needles such as were noticed in the last specimen. Inside this comes a eutectic zone 0.455 mm. thick, followed by a gradation zone 0.780 mm. thick, giving a total penetration of carbon of 2.300 mm.

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 50 SECONDS

125 *Run No. 3, Bar No. 33.* On the outside is a zone of lamellar pearlite containing lines of excess cementite outlining the ferrite crystals as usual, the depth of the zone being 0.325 mm. Inside this comes a zone of eutectic 0.845 mm., followed by a gradation zone 0.975 mm. thick, giving a total penetration of carbon of 2.145 mm.

CASE-CARBONIZING MATERIAL NO. 10

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 55 SECONDS

126 *Run No. 1, Bar No. 22.* On the outside of this specimen is a zone of eutectic approximately 0.975 mm. thick, containing spots here and there of white material probably ferrite, such as were noticed in the other specimens. Inside comes a gradation zone 1.105 mm. thick, giving a total penetration of carbon of 2.080 mm.

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 50 SECONDS

127 *Run No. 2, Bar No. 28.* On the outside is a eutectic zone rather irregular in depth but averaging 0.065 mm., so that the ferrite penetrates to the circumference in some places. Inside comes one of decreasing carbon 1.170 mm. thick, giving a total penetration of carbon of 1.235 mm.

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 15 SECONDS

128 *Run No. 3, Bar No. 34.* On the outside is a rim of pure ferrite 0.065 mm., inside of which is a carbonized zone in which the carbon content decreases to that of the interior. This zone is 0.910 mm. thick, giving a total penetration of carbon of 0.975.

CASE-CARBONIZING MATERIAL NO. 11

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 55 SECONDS

129 *Run No. 1, Bar No. 23.* On the outside of this specimen is a zone of eutectic varying in depth, in some places almost reaching the edge, but of an average depth of 0.195 mm. Inside of this comes a gradation zone 1.300 mm. thick, giving a total penetration of carbon of 1.495 mm.

130 *Run No. 2, Bar No. 29.* This specimen was not carbonized up to the eutectic even on the outside, so that there is simply a zone of decreasing carbon content approximately 1.658 mm., which is the total penetration of carbon.

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 50 SECONDS

131 *Run No. 3, Bar No. 35.* On the outside is a thin rim, maximum thickness 0.007 mm., of ferrite not continuous, but present on most of the circumference. Inside comes a zone of gradation 1.235 mm. thick, giving a total penetration of carbon of 1.242 mm.

CASE-CARBONIZING MATERIAL NO. 12

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 60 SECONDS

132 *Run No. 1, Bar No. 24.* On the outside of this specimen is a rather irregular zone 0.329 mm. thick, of eutectic containing white specks here and there as noted in the other specimens. Inside comes one of gradation 1.235 mm. thick, giving a total penetration of carbon of 1.574 mm.

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 65 SECONDS

133 *Run No. 2, Bar No. 30.* On the outside of this specimen we see a zone of eutectic very irregular and penetrated now and again by ferrite bands. The average thickness of this zone is 0.130 mm. Inside of this comes a gradation zone 1.560 mm. thick, giving a total penetration of carbon of 1.690 mm.

134 *Run No. 3, Bar No. 36.* On the outside is a rim of pure ferrite in some places, in others this is not present. The maximum thickness is 0.007 mm., and inside this comes a gradation zone 1.560 mm. thick, totaling 1.567 mm.

CASE-CARBONIZING MATERIAL NO. 13

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 55 SECONDS

135 *Run No. 1, Bar No. 37.* On the outside of this specimen is a zone of lamellar pearlite containing excess cementite in thin polygonal outlines. The depth of penetration of these outlines varies, but averages 0.325 mm. Inside comes a zone of eutectic 0.520 mm. thick, followed by a gradation zone 1.300 mm., giving a total penetration of carbon of 2.145 mm.

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 70 SECONDS

136 *Run No. 2, Bar No. 39.* On the outside is a zone of eutectic approximately 0.520 mm. thick, but very irregular, in some cases the ferrite globules penetrate almost to the edge. Inside comes a gradation zone of 1.170 mm., giving a total penetration of carbon of 1.690 mm.

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 80 SECONDS

137 *Run No. 3, Bar No. 41.* On the outside is a zone of eutectic averaging 0.291 mm., but very irregular and containing spots of ferrite here and there, penetrated more or less by the ferrite bands from the interior. Inside comes a gradation zone of 1.170 mm. thick, giving a total penetration of carbon of 1.463 mm.

CASE-CARBONIZING MATERIAL NO. 14

ETCHED IN 2.5 PER CENT PICRIC ACID FOR 55 SECONDS

138 *Run No. 1, Bar No. 38.* On the outside of this specimen is a zone of lamellar pearlite containing a great deal of excess cementite, both as polygonal outlines and in a few places as long thick cementite needles. The depth of penetration of this excess cementite varies, but averages 1.040 mm. Inside comes a eutectic zone 0.339 mm. thick, followed by a gradation zone 0.650 mm. thick, giving a total penetration of carbon of 2.029 mm.

139 *Run No. 2, Bar No. 40.* On the outside is an irregular zone of lamellar pearlite containing excess cementite, both as polygonal outlines and large needles, and averaging 1.300 mm. thick. Inside comes a zone of eutectic 0.450 mm. thick, followed by a gradation zone 1.105 mm., giving a total penetration of carbon of 2.855 mm.

140 *Run No. 3, Bar No. 42.* Outer zone of lamellar pearlite containing excess cementite averaging 1.170 mm, both as polygonal outlines and as needles. Inside comes a zone of eutectic of 0.845 mm. thick, followed by a gradation zone of 1.300 mm. giving a total penetration of carbon of 3.315 mm.

DISCUSSION

ALBERT SAUVEUR.[1] Those interested in the case hardening of steel will be greatly indebted to Mr. Lothrop for his extensive investigation of the subject and for the simple manner in which his results and conclusions are recorded. These results on the whole agree closely with our present understanding of the case hardening operation. We have realized for some time that careful heat treatments were needed after carbonizing in order to confer a fine structure both to the core and to the case. It was not so generally known, however, that the strength and toughness of the case-hardened article could be further increased by tempering. It is the writer's opinion that little is to be gained by tempering case-hardened articles for the following reasons:

"There seems to be at first sight no apparent reason why case-hardened articles could not be likewise improved by suitable tempering following the hardening of the case. On second thought, however, it will be realized that since the chief purpose of tempering is to toughen the hardened steel and since hardened articles depend for their toughness on the toughness of their cores little is to be gained by tempering them."[2]

Referring to the exhaustive tests which the author conducted in order to ascertain the relative merits of various carbonizing materials, it is somewhat disappointing that he fails to give the names of the various substances investigated. The reasons for the author's secrecy in this matter are not readily understood. Referring to material No. 14, the writer infers from its composition that it consists of a mixture of charcoal and of barium carbonate and notes that the results obtained by its use are very satisfactory. This confirms his own experience with this kind of material.

J. A. MATHEWS[3] said he had watched the progress of the work described in this paper during the two or three years that it had been going on with a great deal of interest. He had had personally but little to do with the paper, which represented the work not only of Mr. Lothrop and his assistants on the steel making side, but also considerable work of the metallurgical staff of

[1] Prof. of Metallurgy and Metallography, Harvard University, Boston, Mass.

[2] The Metallography of Iron and Steel, Albert Sauveur, 1912, Lesson 16, p. 6.

[3] Genl. Mgr., Halcomb Steel Co., Syracuse, N. Y.

the H. H. Franklin Manufacturing Company and the Brown-Lipe-Chapin Company, both of which had done large amounts of case hardening. The paper was of extreme value, and virtually reduced case carbonizing to a science. By following the directions and observing the results as given in the paper, it would be possible to do case hardening with an extreme degree of accuracy in duplicate parts. That was something that had not always been done. One of the most important points covered by the paper was in relation to the thermal conductivity of the different compounds. That was something that should be known regarding all the different compounds, particularly in the case of changing from one to another where the mere change due to the difference of thermal conductivity of the material in the box would affect the total penetration of the carbon in a given time or at a given temperature. But if the relative conductivities were known such differences could be prevented. Such data had not been available in the past. Mr. Lothrop and his collaborators in this work were entitled to great praise for the thoroughness with which they had carried on this investigation.

Robert R. Abbott.[1] The meaning of the author in many places is obscure. For example, Appendix No. 1, test No. 1, treatment was given as quenching from the pot at 1500 deg. To make a legitimate comparison of this test with tests Nos. 2 to 12 it should have been carbonized in the same furnace and cooled under identical conditions with them. Were the pots removed from the furnace to cool? The carbonized temperature is given at 1700 deg. Then how was this No. 1 test quenched at 1500 deg.? How was this 1500 deg. measured? In the summary at the bottom of Appendix No. 1, Table 12, the best single treatment is given as quenched from 1600 deg. (test No. 4) and yet test No. 1 quenched from pot at 1500 deg. gives a higher rupture load (900 against 850) as well as a better deflection (0.40 in. against 0.24 in.).

In Appendix No. 1, Tables 13 to 21, under Summary 1 is stated "Strength and toughness decreases with increasing depth of case." This is an example of a common error into which many investigators are led. It is a conclusion based upon insufficient or faulty experimental data. The expert investigator eliminates all variables but one. In this case two variables were present,

[1] Metallurgical Engr., The Peerless Motor Car Co., Cleveland, O.

first, a variation in the depth of carbonization and, second, a variation in the *correct* heat treatment, since the different depths of carbon undoubtedly had different percentages of carbon at the surface and the treatment should have been changed to suit the different conditions; instead of this the same temperature was used for all depths. For example, Table 13, Treatment 1: if 1500 deg. produces a certain condition in the case with a depth of 0.01 in., will 1500 deg. produce the same condition when the case is 0.02 in. The writer has not considered this point in drawing his conclusions.

Appendix No. 1, Table 16, Summary 4: The best double treatment is given as 1600 oil—1500 oil, while from the tables this is not as good as 1600 oil—1500 water. In general the writer questions many of the conclusions of the author as given in his summary. For example, comparing Table 16, treatment No. 14 with treatment No. 17, the use of 1500 deg. for the second quench has increased the load from 1240 to 1390, or 12 per cent, while the deflection is decreased from 0.75 in. to 0.44 in. of 41 per cent. There can be no question but that the author has not drawn correct conclusions; 1600 deg. oil—1300 deg. oil is by far the better. A shock test would show this. For the same reason the author's conclusions in Par. 7*f* are incorrect.

In Table 19 the author's best double treatment, namely, 1500—1400 is in reality but a single treatment as the final quench (1400 deg.) is about 20 deg. above the upper critical temperature of the core and therefore both the core and case are in the solid martensitic condition, which is absolutely wrong from a theoretical as well as a practical standpoint.

Considering the author's conclusions regarding the best single and double treatment for different carbon nickel steels:

	Best Single	Best Double	
0.10 Carbon	1500	1600	1500
0.15 Carbon	1500	1600	1500
0.20 Carbon	1450	1600	1400
0.25 Carbon	1500	1550	1400
0.31 Carbon	1400	1500	1400

Why should there be such a drop and then a rise in the best single treatment at 0.20 and 0.25 carbon? Why should the best double treatment have no variation in the best treatment, for the 0.10 and 0.15 carbon and also for the 0.20, 0.25 and 0.31, but vary 100 deg. between the 0.15 and 0.20 carbon? This is manifestly not probable.

In general the value of the experimental data, given in Tables 12 to 21 inclusive, is questionable as this work represents a series of hit and miss treatments with no evidence of a fundamental knowledge of the theory upon which correct heat treatment is based. This is particularly evident in the so-called double treatments, many of which are in reality not a true double treatment, i. e., one in which first the core and then the case is affected.

Particular attention might be called to the fact that many of these treatments could have been improved if the oil and water quenching had not been indiscriminately used. An oil treatment does not affect the case as deep as a water treatment. In other words, an oil treatment, if done at the right temperature, leaves the steel in a condition similar to a steel carbonized to a less depth, but water treated. Briefly then many of the author's results with oil treatment could have been duplicated by a water treatment with a shallower carbonization and therefore with a saving in cost.

The conclusions given in Par. 11, when taken in connection with the temperature given for the best single and double treatments, particularly of steel No. 7, have not been made with due regard for experimental evidence offered. They are incorrect.

Par. 33 *d* is too broad a statement, since the presence of needles of cementite is not caused entirely by high temperature as this can also be produced by a longer time of carbonization at a lower temperature.

The statements in Par. 33 *g* cannot be condemned too strongly in view of what is one of the most important and fundamental operations in the scientific heat treatment of carbonized steel. If the steel has been correctly treated subsequent to carbonization the excess cementite will not exist " in the form of film of carbide between the grains of the metal " as the author states. If the practical man was unable to take advantage of the better wearing surface produced by a carbonization of greater than 1.00 per cent surface carbon without the danger of " rendering it liable to fail by flaking off or spawling " he would certainly be behind the times in these days of rapid increase in the knowledge of heat treating. The above statement also applies to Par. 47 *e*.

The explanation in the second part of Par. 33 *h*, as has been so ably given by Dr. H. M. Howe, is taken by the author as a reason for the flaking off of the case from the core. While this tendency is undoubtedly present if the material is not sub-

sequently given a correct heat treatment, many practical case hardeners can carbonize an open hearth steel at say 1750 deg. in the proper material and then by a correct heat treatment make it as efficient to resist flaking off as could be done by carbonizing at 1650 deg.

In the third part of this paragraph stress is laid upon the coarsening of the grain size by high heat. This is of little practical importance as correct heat treatment will break up any grain produced during the process of carbonization.

Uniform results cannot be obtained by the repeated use of carbonizing material for high-grade work as advocated in Pars. 47 and 48. Mixing part old and part new is not desirable as the old material generally is not uniform, having been used for different lengths of time and at different temperatures.

Benj. E. Bosler[1] presented in a written discussion some notes

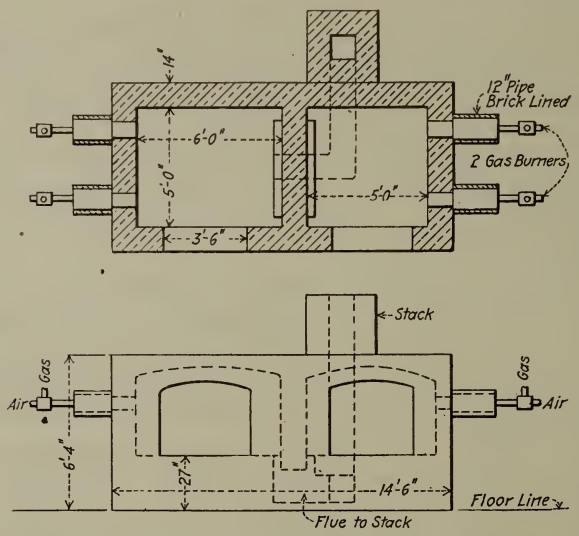

Fig. 31 Furnace for Case Hardening. No. 1 Smith Shop

on case hardening. The parts were case hardened at No. 1 Smith shop. About 700 or 800 bevel and spur gears for mill roller runs have been treated during the past three years, varying in size from 10 in. to 18 in. pitch diameter and about 4 in. to 5 in. width

[1]Asst. Supt. Mech. Dept. Cambria Steel Co., Johnstown, Pa.

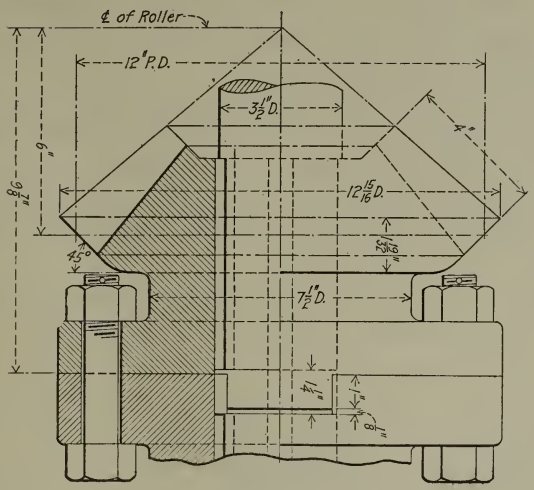

FIG. 32 HALF COUPLING WITH MITRE GEAR, NO. 1 MILL FURNACE RUN; 0.12 PER CENT, CARBON STEEL; CASE CARBONIZED ON FACE AND IN BORE AND KEYWAYS. PENETRATION, $\frac{1}{8}$ IN.

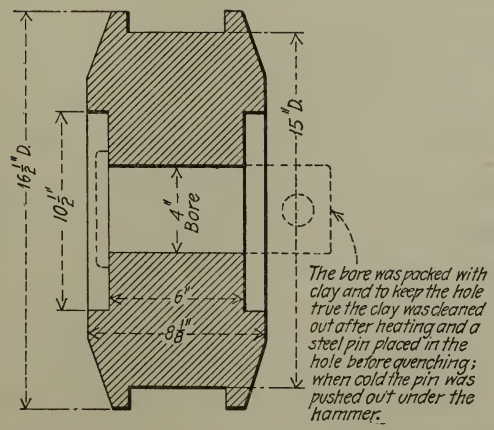

FIG. 33 CASE-HARDENED ROLLER FOR ORE SCREENING PLANT. ABOUT 30 HOURS IS REQUIRED FOR $\frac{3}{16}$ IN. PENETRATION

of face. Practically all crane and motor pinions are now case hardened and a few spur gears as large as 30 in. in diameter by 8 in. face have been treated.

About 90 per cent of the case-hardened gears are made from open hearth steel forged blanks of dead soft stock with carbon between 0.8 and 0.12 per cent. The company has case-hardened steel with a carbon content as high as 0.20 per cent but finds that the best results are obtained by using steel with carbon under 0.12.

The other 10 per cent of the gears treated are steel castings made to the following specifications: carbon 0.10 to 0.15 per

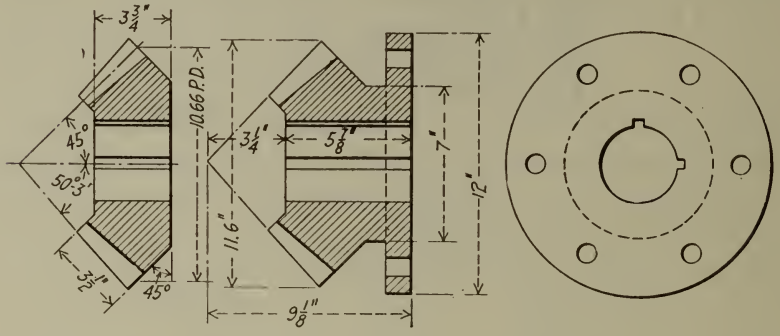

Fig. 34 Gears for 24-in. Mill Tables; Forgings, 0.08 to 0.12 Per Cent, Carbon Steel. About 24 Hours Required for ⅛ in. Penetration

cent; sulphur not over 0.05; phosphorus not over 0.04; silicon 0.15 to 0.30; manganese 0.60 to 0.80.

Regarding the life of the case-hardened gears, the roller run was installed in front of mill furnace No. 1 about four years ago where about 200 case-hardened gears were used and up to this time not one has been replaced, although the table has had very hard service. Gears in the roller runs leading up to the 18-in. mill, in the roller runs at Gautier 24-in. mill and in the tables at the Franklin mills, are giving equally good service.

Among other parts case hardened are bushings for roller bearings (ground after treated); links and chains for conveyor chains (very light penetration, about $^1/_{64}$ in.); and valve motion parts and connecting rod slides for locomotive repairs.

The gears or other parts to be treated are placed in round or square boxes and packed with bone black, using about three-

quarters old to about one-quarter new material. The boxes are then sealed with clay and placed in the furnace where they are heated by natural gas to a temperature of about 1800 deg. fahr. The time required to heat properly varies from 12 hours to 30 hours, depending entirely on the size of the parts being treated and the amount of penetration desired. A great amount of the work requires about 20 to 24 hours from the time the furnace is lighted until the charge is ready to be withdrawn and quenched. This will give about ⅛ in. penetration which is the depth reached on most of the gears.

On light work the charge is usually allowed to cool before it is taken out of the carbonizing material when it is reheated to about 1400 deg. fahr. and quenched in water. Nearly all of the gears and other heavy work are quenched in water direct from the packing box, a method which gives very good results and saves the cost of reheating. The heat in the gear at the time of quenching in this case is also 1400 deg. fahr. as nearly as the operator is able to judge with the eye.

It is found necessary to grind the bores of all gears after hardening as the bore is usually slightly out of round.

Boxes made of 1-in. plate steel will last for about 15 times in the furnace before they become too thin from frequent scaling.

The furnace used is of very simple construction, with two heating chambers each being supplied with two natural gas burners. No trouble whatever is experienced in maintaining an even temperature of 1800 deg. fahr. throughout the time the charge is in the furnace. The time required to bring the cold charge up to this temperature requires from 4 to 6 hours.

A diagram of the furnace used is shown in Fig. 31, and drawings of the pieces case hardened in Figs. 32 to 34.

THE AUTHOR. I desire to thank Mr. Benj. E. Bosler, Dr. Albert Sauveur, Dr. John A. Mathews and Mr. Robert R. Abbott for their kind discussions of my paper.

No. 1373

MEASUREMENT OF AIR IN FAN WORK

By Charles H. Treat, Honor, Mich.

Junior Member of the Society

Air at fan pressures has come to be used in many fields and for widely different lines of work. For the many occasions where measurement of volume, or of pressure and volume, are required better and more accurate methods are often desired than have been commonly used. The engineer frequently needs something better than the ordinary anemometer to determine the velocity, and a rubber tube connected to a water gage to get the pressure. Especially is this so in fan testing where the volume varies from nothing with closed gate to a maximum with free discharge. The measuring apparatus must be accurate through this wide range.

2 The pitot tube[1] used with suitable pressure measuring devices is particularly well adapted for measuring both pressure and volume, being easily and conveniently arranged for fan testing, also being simple, reliable and accurate.

3 Although the pitot tube has long been known and used there is still much doubt of its accuracy and distrust of results obtained by it. This is especially so for air. The writer believes the trouble in some cases and the resulting general distrust are due to a lack of information as to what forms can *not* be used. In this paper are described what is believed to be an accurate form of tube for this purpose and a sufficiently simple way of using it. The pitot tube will be the only means[2] described, although much of the method employed in the test can be used however the values of pressure and volume were found.

4 This pitot tube work was done from time to time in the Amer-

[1]For references and a good account of what has been done see The Pitot Tube, W. B. Gregory, Trans. Am. Soc. M. E., vol. 25, p. 184.

[2]Professor Thomas of the University of Wisconsin has developed an excellent device for measuring air or gases known as the Thomas Meter.

Presented at the Annual Meeting 1912, of The American Society of Mechanical Engineers.

ican Blower Company's fan testing room in Detroit. A little turbine fan was being experimented with by the writer who wanted something smaller and better adapted for use in the small test pipe than the form of tube previously used.[1] The form of pitot tube adopted and which has been in almost daily use for a number of years is shown in Fig. 1. This has been tried out in many ways as by measuring the same air through pipes of different diameter; by measuring in the same pipe with two or more instruments; by measuring the air displaced by a gasometer. These three checks bring in also the

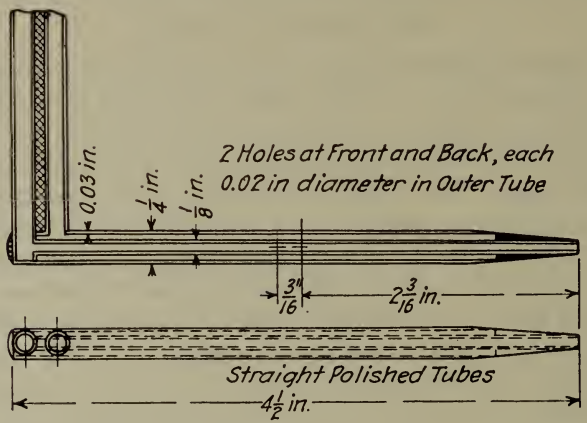

FIG. 1 STANDARD PITOT TUBE

important factor in air measuring of getting the mean velocity in the pipe.

CALIBRATION OF IMPACT AND STATIC ORIFICES

5 In deciding how experimentally to prove the correctness of the orifices of the pitot tube, preference was given to the method which seemed most simple and accurate and which promised to be the best check on work already done.

6 The accuracy of the impact orifice can not be proved by means of a jet of air because the only practical means for determining the velocity at any point in the jet is by the use of the pitot tube. It would scarcely do to test the tube by means of a carrier, such as a locomotive, because of changing air currents. But by attaching the impact tip to the end of a slender hollow arm rotating about a shaft, uniform motion may be obtained and that in air having motion due only to the slender arm through it. If now the shaft be a fan

[1]The pitot tube previously used was the static orifice shown in Fig. 11 at *D*, used with a $\frac{5}{16}$-in. tube pointing into the air current for impact orifice.

shaft and the fan be discharging, air is drawn to the inlet about which the arm is rotating, and by regulating the fan discharge, air can be just drawn away from the path of the tip or, in other words, air flow will be towards the inlet and at right angles to the tip travel instead of with it.

7 We have here a simple means to compare the pressure computed from the relative velocity of tip to air, by the relation $v = \sqrt{2hg}$,

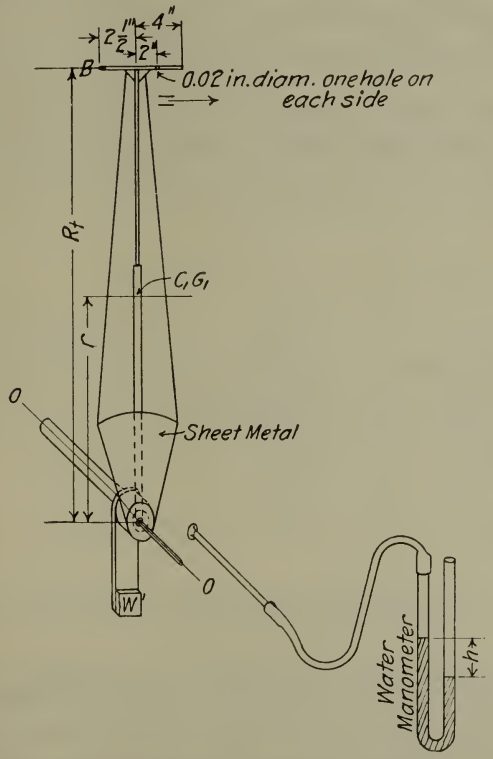

Fig. 2 Diagram for Apparatus for Calibrating Pitot Tube

with the pressure actually resulting from this velocity. We deal with pressure due to impact only as the atmospheric pressure in the room is on the tip and pressure measuring instrument alike.

8 This rotating arm method was the one used for calibrating not only the impact orifice but the static orifice as well. For the static orifice what we wish to know is whether or not it shows the true pressure of the fluid no matter what the velocity may be. As there may be slight radial flow static openings had best be put at the sides

(Fig. 2) for then this velocity is only a component of the total velocity relative to the static opening. If the pressure in the tip be the same as that in the room the pressure at the axis will be less by the centrifugal head in the arm.

9 If the fan be not used the flow caused by the rotating arm is enough to reduce the pressure in the tip from about 2 to 5 per cent, while the component of flow toward the inlet when the fan is used even with the tip pointing to take the resultant maximum velocity will change the pressure less than will show on a slide rule, flow towards the inlet being only enough to take the air away.

10 In older pitot tube work total or impact reading has usually been all right, but static readings have in many cases been inaccurate. A very simple and reliable check on static openings, but which may not be quite accurate, is to try them in a jet of air blowing out freely

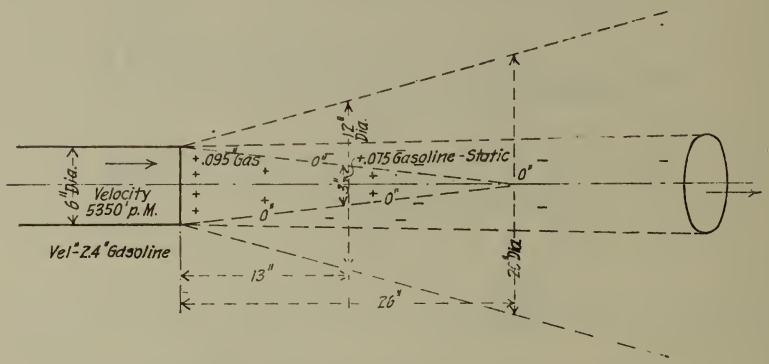

Fig. 3 Static Pressure at Outlet of a Straight Pipe

into the room and observe if the pressure is the same as that in the room, as it should be very nearly even with high velocity. Fig. 3 shows what the static pressures were at the end of a straight 6-in. diameter pipe, the velocity pressure being 2.4 in. head of gasolene, equal to 1.74 in. of water.

11 For details of the calibration work, method of computing the centrifugal head, data of tests, etc., see the Appendix. The results are as follows: Working in air instead of water, it is shown that there is no difficulty in getting correct impact or total readings. The form of the tip may be changed from a straight open tube pointing into the current to a small pointed orifice, as in Fig. 1, or may be flat at the tip with a small orifice in the center and still show practically no difference in the readings for the different shapes, showing

that for any ordinary change in the tip no inaccuracy results. The form of impact tip most carefully checked was that shown in Fig. 1.

12 The calibration shows the impact or total reading to be accurate probably to within 1 per cent in the relation $v = \sqrt{2gh}$.

13 A correctly reading static orifice is not so easily obtained. The static of Fig. 1, consisting of clean holes 0.02 in. in diameter in $\frac{1}{4}$-in. tubing $\frac{1}{32}$ in. thick, gave static readings accurate to within less than 1 per cent of pressure due to velocity. A hole $\frac{1}{16}$ in. in diameter in this tubing gave readings considerably off, while a $2\frac{1}{2}$-in. slot $\frac{1}{16}$ in. wide in slightly larger tubing gave static readings as much as 14 per cent too high.

14 An older form of static orifice, shown at *D* in Fig. 4, is fairly accurate, but requires to be set with the plate exactly parallel with

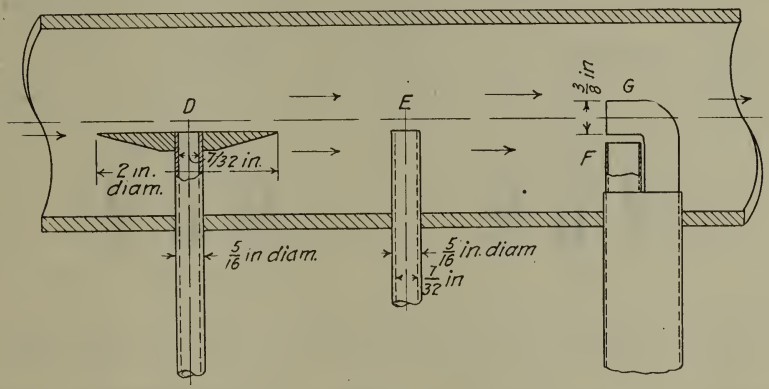

Fig. 4 Older Forms of Static Tips

the direction of flow. It is often not known just what the direction of flow is, as when there is a slight whirling or spiral motion in the duct. The static openings of the pitot tube shown in Fig. 1 give quite accurate results when 5 deg. out of parallel with flow.

15 The open tube *E*, Fig. 4, gives static readings too small by over 50 per cent of the velocity reading. The tube is $\frac{5}{16}$ in. in diameter held at right angles to flow. The static tube *F* in a form of pitot tube that was sold by one of the instrument makers several years ago gives static readings too small by 65 per cent of the true velocity reading. The impact tube gives practically true reading, so that velocity readings by this instrument are much too large.

16 A $\frac{5}{8}$-in. slot 0.01 in. or less wide in the $\frac{1}{4}$-in. tube seemed to give good results with or without inner tube, so also did the large

slots if an inner tube fitted the outer to within 0.01 in. on each side. Wire cloth over the large slots seemed to give good results. These other forms of opening have not been checked and rechecked as have the small 0.02-in. holes. Where the small holes are liable to stoppage, more holes should be used.

METHOD OF TESTING FANS

17 When air is discharged into a pipe by a fan both static and velocity pressures measured from atmospheric are positive, and their sum is shown by the reading in the impact manometer (*B* in Fig. 5). When the air is drawn from a pipe by a fan the static pressure in the pipe is always less than atmospheric; that is, it is negative. The reading in the impact manometer *B* will be also

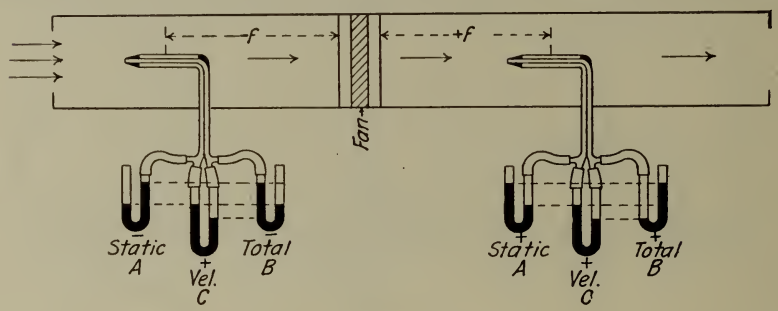

FIG. 5 ARRANGEMENT FOR MEASURING STATIC AND VELOCITY PRESSURES

negative, but less than static. Static subtracted from the impact or total gives a positive reading, as it should, for the velocity. As velocity in this pipe is induced by the static pressure, the velocity can never quite equal the static due to the presence of friction.

18 Figs. 6 and 7 show the method of setting up for a fan test: Fig. 6 for a test on the suction and Fig. 7 for a test on the discharge. Fig. 8 is from a photograph showing the actual arrangement of apparatus under test.

19 By using the pitot tube shown in Fig. 1, where velocity and static are measured practically at the same point, and piping as at *C*, Fig. 5, the pressures other than that due to velocity cancel each other. The possibility of the pressure changing between times of reading static and impact independently is avoided and there is no subtraction to be made. The difference in static pressure between the point of reading in the pipe and atmospheric pressure is read at

A, the total at *B*. The pitot tube should point straight into the current of air to read correctly, either velocity or static.

20 The common pressure measuring apparatus for pressures above 1 in. water gage is satisfactory. For pressures from 1 in. to 2 or 3 ft. of water, manometers are reliable and convenient. Various forms of apparatus are used for measuring pressures of 1 in. or less. As simple and reliable a means as any is probably furnished by an inclined manometer containing gasolene instead of water. The tubes for this manometer should be of the same size, perfectly straight and parallel inside. When so made they can be easily used when inclined as much as 20 in. slant for 1 in. vertical reading, with very accurate results. Water is not a satisfactory fluid in a manometer inclined more than three to one. Gasolene is recommended, as it is not only very active but comes accurately to the same form of surface.

21 Set the back plate by leveling by bubbles in the manometer tubes and then swing the tubes up the angle required to give a good reading, 20 or 10 or 5 to 1. In fan testing for velocity reading at different restrictions two or three slants may be used to cover the wide range in pressure and it is very convenient to cover this wide range with the one gage by the simple device of changing the angle. Good care is necessary to get the angle correct for 20 to 1 and great care for 40 to 1.

22 At 40 to 1 slant for $\frac{1}{2}$ in. velocity reading the velocity is 384 ft., for 0.49 in. reading the velocity is 380 ft., showing that with the slant gage velocities as low as 500 or even 400 ft. per min. may be measured satisfactorily.

23 With skill and care the hook gage gives good results if the pressures do not fluctuate. In air measuring work the pressures usually fluctuate slightly even when care is taken to get the speed of fan as constant as possible. It is desirable to be able to take the readings under these conditions, and this may be done by setting zero-point of the measuring slide at the mean lower level, and at the instant the level comes opposite this point, reading at the upper level.

24 The spacing board for the pitot tube should be placed so that the tube crosses the test pipe centrally. On this board lay off *AB* at the center of travel of the holding block *K* for the tip at extreme movement across pipe. Compute *a*, *b*, *c*, *d* and lay off with *AB* at their center. Inclining manometers are shown for pressure measuring, but other instruments for this purpose may be used if

preferred. After connecting try out the lines of tubing by noting if the liquid lowers in the manometers after pressure has been applied, and make sure that the connections are not clogged. Use care to prevent liquid from being drawn from the gages into the tubing.

25 Make the area of the test pipe the same as the area of the outlet or inlet connections for a fan test. Use well fitted soldered pipe for the test. An ordinary test pipe unsoldered and somewhat carelessly fitted to the fan may leak 5 per cent or more of the air

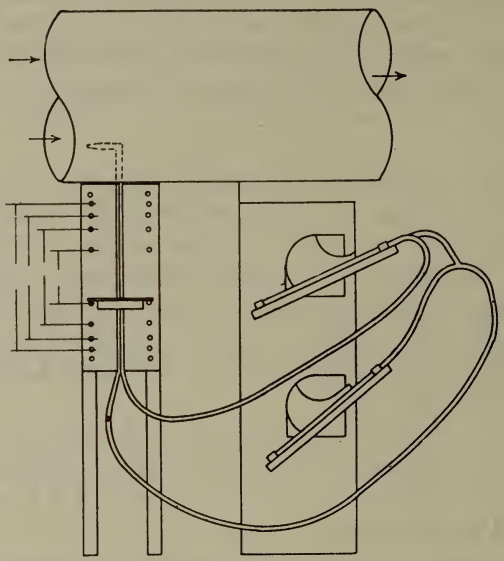

FIG. 6 CORRECTION FOR TEST ON SUCTION

handled at $\frac{1}{2}$ gate, or even at $\frac{5}{8}$ gate, lowering the pressure correspondingly.

26 It is very essential to determine the relation between volume, the pressures and the power required as the resistance changes. For this purpose it is usual to gate down the outlet for an outlet test or the inlet for an inlet test, using round orifices in thin flat plate. No air should escape except through the orifice. It is usually ample to test with gates 0, $\frac{1}{4}$, $\frac{3}{8}$, $\frac{1}{2}$, $\frac{5}{8}$, $\frac{3}{4}$, and once the area of the orifice. Occasionally a $\frac{7}{8}$ gate is useful. For some tests the $\frac{3}{8}$ and the $\frac{5}{8}$ gates may be omitted. The area of the gate divided by the area of the test pipe is the ratio of opening or the equivalent orifice.

27 The coefficient of efflux is larger for a conical outlet than for an orifice of the same area and is probably greater for a large gate that for a small on the same test pipe. However, a gate, say $\frac{3}{4}$, is very nearly a fixed measure of resistance as above made. Even if it were not there would be no inaccuracy due to this cause as the final curves are ordinarily used, for the *required* ratio of opening is found from the *required* ratio of pressures. The ratio of gate area to full area is often very useful, so that it is wiser to use ratio of opening for abscissa in final curves than some other quantity

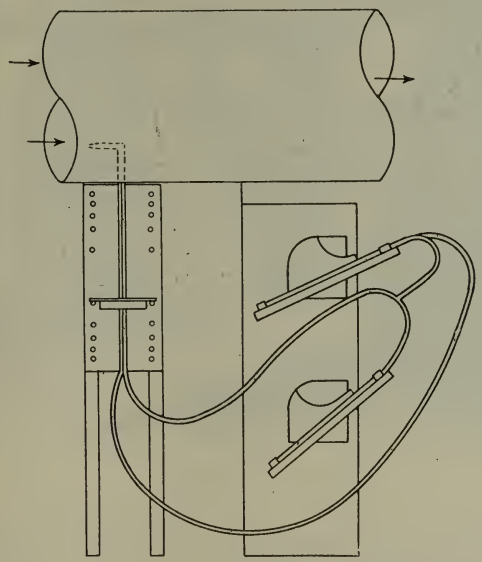

FIG. 7 CORRECTION FOR TEST ON DISCHARGE

as volume or ratio of $\frac{V.P.}{D.P.}$

28 A convenient log sheet for fan testing is shown in Fig. 9 On this log and on the curve sheets, Figs. 9 and 10, pressures are all expressed in inches of water. The pressure in the impact tube or the total or dynamic pressure is marked Dyn. *W. G.* or simply *D.P.* The pressure in the static orifice (*A* in Fig. 5) is Static *W.G.* or simply *S.P.* The difference between the static and the total (*C* in Fig. 5) is the velocity pressure and is marked Vel. *W.G.* or simply *V.P.* The pressure due to a velocity of the air equaling the peripheral velocity of the wheel is the peripheral velocity pressure or *P.V.P.*

29 In making fan tests the speed may not be constant at all times. To correct to a common speed we have relations between speed and pressure and power that have been fully verified by test, the better the tests the better being the check. See curve sheet *A* for a good test showing this point. These relations are reasonable and could probably be established by theory. They are as follows:

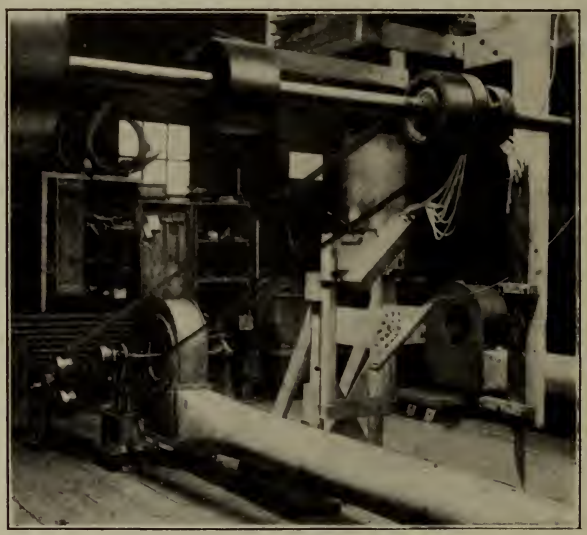

FIG. 8 FAN TESTING

30 Under the same conditions of resistance to air flow, density of air and with the same fan

a The volume of air discharged by the fan varies as the r.p.m., that is

$$\frac{\text{vol.}}{\text{Vol.}} = \frac{\text{r.p.m.}}{\text{R.P.M.}}$$

b The pressures produced vary as the square of the r.p.m.

$$\frac{\text{static}''}{\text{STATIC}''} = \frac{\text{total}''}{\text{TOTAL}''} = \frac{\text{velocity}''}{\text{VELOCITY}''} = \frac{\text{r.p.m.}^2}{\text{R.P.M.}^2}$$

c The horsepower to drive varies as the cube of the r.p.m.

$$\frac{\text{h.p.}}{\text{H.P.}} = \frac{\text{r.p.m.}^3}{\text{R.P.M.}^3}$$

31 *Friction in Test Pipe.* In working up a fan test, a correction for friction in the test pipe must be made to show what the fan alone

FIG. 9 LOG SHEET FOR FAN TEST

is really doing. The friction pressure or pressure to overcome this friction may be 10 to 15 per cent of the total at wide open gate and 3 to 4 per cent at ½ gate.

32 For ordinary round galvanized pipe a simple and quick way to find this friction is from the formula

$$F = \frac{\text{Vel.}'' \times L'}{40 \times D'} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

where F = friction in inches of water L and D = length and diameter of round pipe. Or in a form more easily remembered, the length of pipe in which friction inches equal velocity inches pressure is 40 diameters long. This rule is applicable for air under ordinary conditions of temperature, pressure and humidity, and values by it check closely with those given by Kent.

33 For smooth straight pipe instead of 40 times the diameter it may be 60 or more and for rough beaded pipe the figure will be less that 40.

34 A general equation[1] for friction is

$$P = \frac{fsyv}{A2g}$$

for any pipe, and

$$P = \frac{4\,fyLv}{2gD} \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

for round pipe

y = weight of 1 cu. ft. of air
P = pressure lb. per sq. ft.
L and D = length and diameter of pipe in ft.
f = coefficient of friction
v = velocity in ft. per sec.
s = perimeter of section of pipe

41 If we substitute in [2] the value 0.0067, that being the coefficient of friction in ordinary piping, for coefficient of friction and 0.0715 as the weight of air we can derive [1]. It is interesting to note that 0.0067 is given as the coefficient of skin friction for water of rather rough surface. Is the coefficient for air and water the same? Why should they not be the same?

42 *Standard Weight Air.* It is important to have a standard weight of air at which fan performances may be compared. The density, while ordinarily a small factor, affects the results consider-

[1] See Church's Hydraulics.

ably, if extremes are great, as with heated air or with cold out-door air. It affects the velocity pressure, the total pressure and the horsepower.

43 As correction to a standard involves some labor it is desirable to choose for a standard a mean weight so that the original data will be close enough without correction in many cases, or so that the conditions may be kept such as will give standard weight. Another reason for the mean or ordinary weight is that such data as taken may be compared with previous tests, also that ordinary rough uncorrected tests may be compared with standard tests.

44 As fan tests are apt to be conducted indoors with warmed air the weight on the average for the year in the United States will not be far from 0.0715 lb. per cu. ft. This in fact is the average weight for a large number of tests extending over a number of years. This weight gives a velocity constant of 4100 or

$$\text{vel.} = 4100 \sqrt{\text{vel.}''}$$

There is some advantage in a short figure in computations from the final curves.

45 To correct to standard air, multiply the velocity pressure, total pressure and the horsepower of the actual test by $\frac{0.0715}{\text{wt. 1 cu. ft.}}$

The velocity will be the same as though found at the actual velocity pressure, but the static and total pressure change as the density, changing the air horsepower and the actual horsepower in proportion.

45 A fan discharging under constant conditions as to speed and resistance, will maintain a constant head and a constant velocity of discharge with changing density. That this is so is apparent from the following:

47 In a pipe system supplied by a fan running at a constant speed all the resistance may be classed as friction of various kinds. The impeller sets up heads, due to centrifugal force and to motion imparted to the air, which total head is resisted by internal and pipe frictions. The net head

$$h_e + h_v + h_f + h_{fi} = \frac{V^2}{2g} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

48 Density of the fluid does not affect these heads, and the net head for one density will be the head for all. The velocity will be the same for all from the law of falling bodies.

Centrifugal head

$$P = h_c y = \frac{wv^2}{gr} = \frac{2ryv^2}{gr} \quad \ldots\ldots [4]$$

$$h_c = \frac{2rv^2}{gr} \quad \ldots\ldots [5]$$

Head due to motion

$$h_v = \frac{v^2}{2g} \quad \ldots\ldots [6]$$

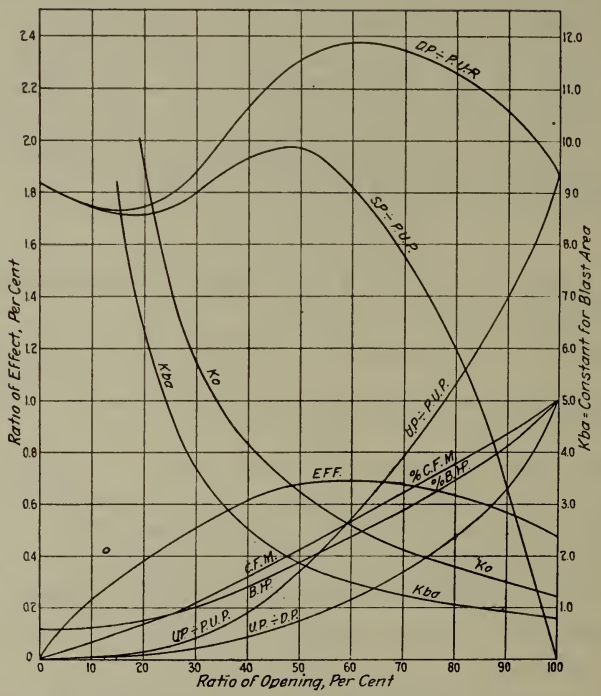

Fig. 10 Characteristic Curves of a Fan Test

Heads due to friction pressure

$$P = h_f = \frac{fSyv^2}{A2g} \quad \ldots\ldots [7]$$

and

$$h_f \text{ or } h_{fl} = \frac{fSv^2}{A2g} \quad \ldots\ldots [8]$$

h_c = head due to centrifugal force
h_v = head due to velocity and velocity change

h_t = head due to friction in case
h_{ft} = head due to friction in pipe
P = pressure per sq. ft.
y = density or weight of 1 cu. ft. air

49 From this it follows that the volume discharged from a fan is the same at all densities, other conditions being the same. This is probably true for all fluids, since the coefficient of friction seems to be the same.

50 A thoroughly good way of making the results of a fan test available and useful is to plot characteristic curves, as in Fig. 10.

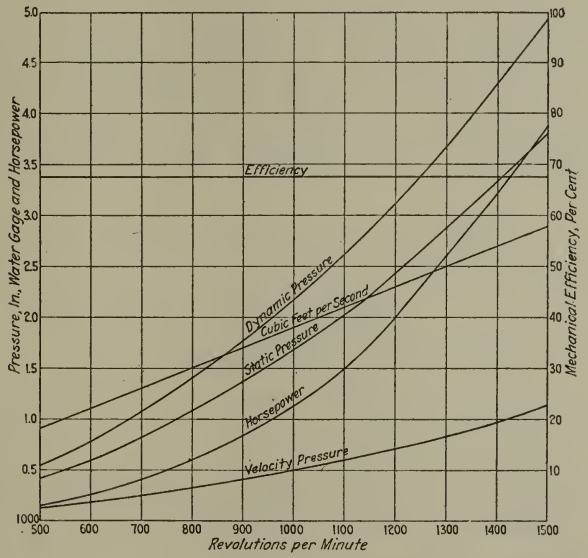

Fig. 11 Preliminary Curves of Fan Test

51 The original readings are first corrected to one common speed, if the speed has varied through the test, as it is likely to do from belt slip. These data at a common speed are then plotted on a preliminary curve sheet similar to the curve sheet in Fig. 11, but with the ratio of opening plotted as abscissa instead of r.p.m. On this sheet correction can be made to standard weight air and for pipe friction. This preliminary set of curves will usually show up a mistake in observation or in computation. From this curve sheet practically any fan problem may be solved for fans of that kind.

Of course the other fans of the line must be kept in proportion to the one tested to get the result.

FORMULAE AND CALCULATIONS USED IN TESTS

52 Bernouilli's theorem states that the sum of all the pressures of a fluid at one section of a system of smooth piping equals the sum of all the pressures at any other section, pipe friction being cared for by the difference in friction pressures.

53 If a fluid motor be introduced between the two sections where pressures are taken it will create a difference in pressure. This difference in pressure times the volume of fluid passing through the motor times the constants for heaviness of fluid, area, etc., gives the horsepower produced by the motor, or given to it if power is being taken from the fluid. But for the pressure caused by the motor, the sums of pressures at the two sections would be equal:

$$\frac{\text{cu. ft.}}{\text{const.}}\left[-\Big(v''+(-s''-f'')\Big)+(V''+S''+F'')\right]=\text{h.p.}$$

$$\frac{\text{cu. ft.}}{\text{const.}}\Big[V''+S''+F''+s''+f''-v''\Big]=\text{h.p.}\quad\ldots\ldots\ldots\ldots[9]$$

$$\frac{\text{cu. ft.}\times\text{difference in pressure}\times 62.3}{33000\qquad\qquad 12}=\text{air h.p.}$$

$$\frac{\text{cu. ft.}\times\text{difference in pressure}}{6356}=\text{air h.p.}$$

v'' = velocity, in. suction
s'' = static, in. of water
f'' = friction, in. from fan inlet to measurement on suction
V'' = velocity, in. water in discharge
S'' = static, in. water in discharge
F'' = friction, in. at discharge from point of measurement to fan (see Fig. 5)

When the friction heads F'' and f'' are considered as belonging to the air motor, then

$$\frac{\text{cu. ft.}}{\text{const.}}\Big[V''+S''+s''-v''\Big]=\text{air h.p.}\quad\ldots\ldots\ldots\ldots[10]$$

54 When the discharge pipe and suction pipe are of the same diameter $V''-v''=0$ and equation [10] becomes

$$\frac{\text{cu. ft.}}{\text{const.}}\Big[S''+s''\Big]=\text{air h.p.}\quad\ldots\ldots\ldots\ldots\ldots[11]$$

55 It is quite usual in buying and selling fans to understand that a certain effect on discharge shall be produced; that is, such a volume and pressure in discharge pipe.

56 When the work sold for is to be done on the discharge side the inevitable friction loss in drawing air through the inlets is not to be measured in. It is not wanted although as in the case of some single inlet fans it may be a very appreciable factor. Large inlets reduce it. For this case

$$\frac{\text{cu. ft.}}{\text{const.}}\left[V''+S''+F''\right]=\text{air h.p. on discharge side}\ldots.[12]$$

and

$$\frac{\text{air h.p.}}{\text{h.p. to fan}}=\text{efficiency for discharge side}$$

For a fan that does its work on the suction side with a short outlet on the discharge the same size as the inlet, a usual arrangement, we have

$$\frac{\text{cu. ft.}}{\text{const.}}\left[V+0+0+s''+f''-v''\right]=\text{air h.p. from [9], or when } V''=v''..[13]$$

$$\frac{\text{cu. ft.}}{\text{const.}}\left[s''+f''\right]=\text{air h.p. on suction side}\ldots\ldots\ldots[14]$$

If it is agreed to use an expanding or enlarged outlet such as the chimney of mine fans, equation [13] with V'' due to velocity at the outlet of the chimney should be used.

57 It might be agreed to sell for work on the suction side alone, then $\frac{\text{cu. ft.}}{\text{const.}}\left[s''+f''-v''\right]=$ air h.p. and it would be necessary for the fan maker to reduce V'' as much as possible.

58 When a fan is to do work on both sides the equations [9], [10] and [11] ought to be used to measure the air h.p.

59 A cone on outlet will not increase the efficiency of a fan but will decrease work to be done, that is, the same volume of air can be moved through the same resistance by less power, the speed being a little slower.

60 For air measuring work the formula $v=\sqrt{2gh}$ may be written

$$V=1097\sqrt{\frac{\text{vel.}''\text{ W. G.}}{\text{wt. cu. ft. air}}}\ldots\ldots\ldots\ldots\ldots[15]$$

When head due to velocity is expressed in its equivalent inches water gage readings, and V = Vel. in ft. per min.

$V = 4100 \sqrt{V'' \text{ W.G.}}$ for standard weight air of 0.0715 lb. per cu. ft.

AVERAGE VELOCITY OF FLOW IN A PIPE

61 The velocity is least at the surface of the pipe because of skin friction. In a square pipe the mean of the velocities and of the

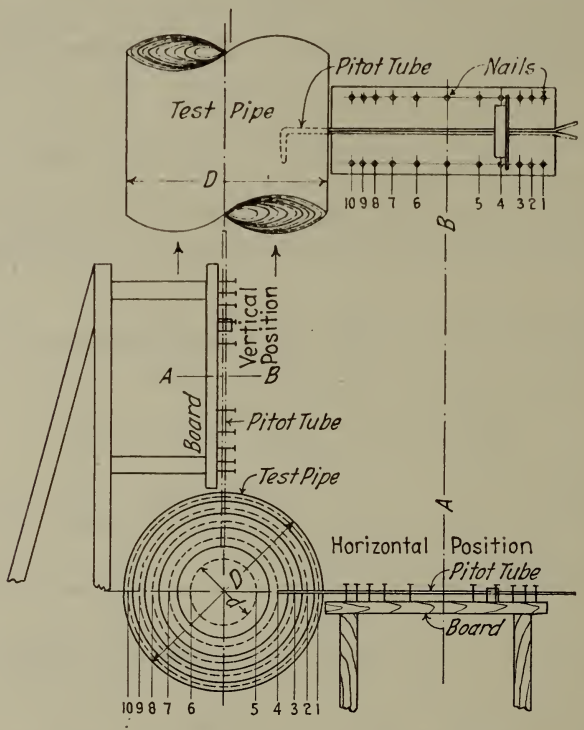

Fig. 12 Setting Pitot Tube

pressures taken at the center of a large number of equal squares into which the pipe is divided, is the average velocity and pressure in the pipe. In the case of a round pipe, divide into equal annular areas, and for each read at a point where the area inside equals the area outside (see Fig. 12). A convenient rule for obtaining diameters of circles is as follows:

62 Divide the diameter of the air pipe by the square root of twice the number of readings to be made from the circumference

to the center for the diameter of the smallest circle. Multiply this reading by $\sqrt{3}$, $\sqrt{5}$, etc., for diameters of larger circles. For example, for $48\frac{7}{16}$ diameter of pipe and eight readings $\frac{48\frac{7}{16}}{\sqrt{2\times 4}} = 17\frac{1}{8}\times\sqrt{3}$, $\sqrt{5}$, etc. $= 29\frac{5}{8}$, $38\frac{1}{4}$, etc. This is done very readily on the slide rule by dividing the diameter pipe found on the lower scale by the square

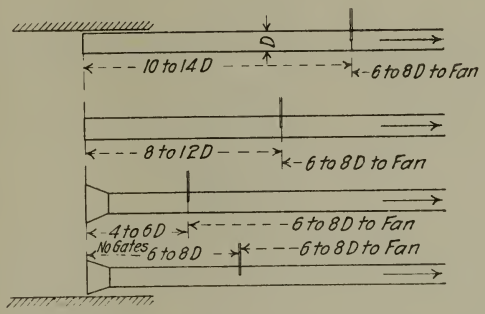

FIG. 13 LENGTH FOR TEST PIPE ON SUCTION

TEST IN DISCHARGE PIPE 20 DIAMETERS LONG. READINGS 5/8 IN. LENGTH FROM FAN

Position of pitot tube across pipe..	1	2	3	4	5	6	7	8	9	10
Velocity pressure in test pipe, in. .	0.730	0.885	0.990	1.025	1.060	1.075	1.055	1.022	0.925	0.680
Velocity, ft. per min.....	3503	3850	4080	4150	4222	4260	4220	4150	3950	3380

Mean velocity pressure in test pipe, 0.9447 in.; mean velocity, 3976.5 ft. per min.
$4100\times\sqrt{0.9447}$ in. $= 3985$ ft.

root of the numbers found on the slide. Without moving the slide multiply by $\sqrt{3}$, $\sqrt{5}$, etc., which roots are also found on the slide.

63 This rule may be demonstrated as follows: Divide the annular rings in two by a vertical line through the center. (See Fig. 12.)

$$\text{Total area} = A = \frac{\pi}{4}D^2$$

Circle d contains two of the equal areas of the 20 into which D is divided.

$$\text{a} = \text{area of small circle} = \frac{\pi d^2}{4}$$

$$\text{Unit area } \frac{a}{2} = \frac{A}{20}$$

$$A = \frac{a}{2} \times 20 = 10a$$

$$10d^2 = D^2$$

since the area of a circle varies as the square of the diameter.

$$d^2 = \frac{D^2}{10}$$

$$d = \frac{D}{\sqrt{10}}$$

64 Since in equation [15] the velocity changes as the square root of the velocity water gage the velocities should be computed for each unit area and those averaged. The error is small, however, if the velocity be computed from the average of the velocity inches, ordinarily being less than 1 per cent from this cause.

65 There are usually irregularities of flow in air entering a pipe and if it is the discharge from a fan the irregularities may be considerable; if an inlet pipe there will be less and if fitted with cone inlet there will be considerably less irregularity of flow. This irregularity becomes less away from the pipe inlet and in 8 to 12 diameters in the case of fan discharge and in straight pipe good pitot tube readings may be taken. (See Fig. 13.)

66 For accurate measurement it is recommended that both vertical and horizontal readings across the pipe be taken rather than either alone, especially if there has been a bend in the pipe or if the pipe is the discharge from a fan.

APPENDIX

CALIBRATION OF STATIC AND IMPACT ORIFICES

67 This calibration was made by revolving the various tips at the end of a slender arm and noting the speed and the pressures at the axis. The arm was made of brass tubing $\frac{9}{16}$ in. and $\frac{3}{8}$ in. in diameter, sweated together. Stay wires were used to give it rigidity, as shown in Fig. 2, and the tubing was fastened to its countershaft by the set screw of the collar through which the larger tube fastens. The different tips tried were soldered to the end of this arm. A very small air-tight connection or stuffing box at the axis was afforded by a $\frac{1}{8}$-in. brass tube projecting from the larger tube along the axis and having a ring of greased candle wicking on it against which the cone end of a larger tube was pressed, the small tube revolving in it. The arm was counterweighted at W. Care was taken to press on the outer tube only hard enough to keep the connection tight.

68 The revolutions were counted by a hand-speed counter at the other end of the shaft, usually for two or three minutes. The length of arm was measured from axis to center of orifice by steel scales. The shaft was driven by an automatic engine having no other load so that the speed was very steady and uniform.

69 Pressure was taken by a vertical water manometer of large tube area (0.45 in. inside diameter), in most of the runs. An inclined gasolene manometer was used, in those of small head the reading being corrected to the head of water. Temperature and humidity were taken in the room not far from the arm.

70 The $\frac{1}{4}$-in. tubes at the end of the arm which were used in all the runs except the last four projected forward from the arm about $3\frac{1}{4}$ in. and backward about $2\frac{1}{2}$ in. The static holes in the sides were about 2 in. ahead of the arm. Different tubes were used but were adjusted alike, being soldered to the corner braces and pipe.

71 Air being elastic, that portion nearer to the shaft presses, due to centrifugal force, on the air further out in the tube compressing it slightly. Therefore the outer portion of the air in the tube is slightly more dense than the inner, and this varying density makes it necessary to go to calculus for a solution.

72 For an inelastic fluid the computation would be very much simpler, and for a rotating column of unit area it would be as follows:

$$\text{Centrifugal force (pressure per sq. ft.)} = \frac{wv^2}{gr} = \frac{w(\pi 2rn)^2}{gr} = \frac{2ry(\pi 2rn)^2}{gr}$$

w = total weight of fluid in lb.
v = velocity in ft. per sec.
g = gravity
r = radius to center of gravity
y = weight per cu. ft. of fluid
n = revolutions per second

DIFFERENCE IN PRESSURE BETWEEN THE ENDS OF A REVOLVING RADIUS OF ELASTIC FLUID

73 Assume
PV = constant
p = pressure in lb. per sq. ft. at any intermediate point, variable
P_o = pressure in lb. per sq. ft. at axis due to the atmospheric pressure
P_t = pressure in lb. per sq. ft. at the free end of the radius—at tip
W = weight per cu. ft. of elastic fluid, variable
W_o = weight per cu. ft. of elastic fluid at atmospheric pressure
N = r.p.m.
R = radius in ft.
R_o = minimum radius
R_t = maximum radius
$B\alpha$ = barometer at time of experiment

74 For a very short radius of elastic fluid

$$p = \frac{WR\left(2\pi\frac{RN}{2\ 60}\right)^2}{g \times \frac{R}{2}} = \frac{2\pi^2 WR^2N^2}{g \times 3600}$$

$$dp = \frac{2 \times 2\pi^2 WN^2}{g \times 3600} RdR$$

$$\frac{W}{p} = \frac{W_o}{62.3 \times \frac{B\alpha \times 13.61}{12}}$$

$$dp = \frac{4\pi^2 \times 12W_oN^2pRdR}{g \times 3600 \times 62.3 \times 13.61 \times B\alpha} = C \times pRdR$$

$$\frac{dp}{p} = CRdR$$

$$\int_{P_o}^{P_t} \frac{dp}{p} = C \int_{R_o}^{R_t} RdR$$

integrating

$$\log_e \frac{P_t}{P_o} = \frac{C}{2}(R_t^2 - R_o^2) + K$$

when $R_t = R_o$, $P_t = P_o$ and $\frac{P_t}{P_o} = 1$. Log of 1 = o and for this $(R_t^2 - R_o^2) = o$, K must = o. As $\log_e$ = common log $\times$ 2.302585 we have

$$\text{common log } \frac{P_t}{P_o} = \frac{C}{2 \times 2.302585}(R_t^2 - R_o^2) \ldots\ldots\ldots\ldots [18]$$

$\frac{P_t}{P_e}$ is the ratio of pressure at the ends of radius and may be expressed in any units of pressure or head.

$$\frac{C}{2 \times 2.302585} = \frac{2 \times \pi^2 \times 12 \times W_o \times N^2}{2.302585 \times g \times 3600 \times 62.3 \times 13.61 \times B\alpha} = 0.0000010479 \frac{W_o \times N^2}{B\alpha}$$

No.	log	No.	log
2	0.301030	2.302585	0.362210
π^2	0.4971498	32.16	1.507316
	0.4971598	3600.	3.556303
12	1.079181	62.3	1.794488
	2.3745106	13.61	1.133858
	8.354175		8.354175

0.0000010479.....6.0203356

Equation [18] with numerical constants is

$$\text{common log } \frac{P_t}{P_o} = \log \frac{\text{in. mercury}}{B\alpha} = 0.0000010479 \frac{W_o N^2 R_t^2}{B\alpha}$$

75 *Impact Orifice.* When the pressure at the axis is atmospheric, or zero on the manometer then the common log of pressure at the tip in inches of mercury is

$$0.0000010479 \frac{W_o N^2 R_t^2}{B\alpha} + \log B\alpha \ldots\ldots\ldots\ldots\ldots [19]$$

The number corresponding to log $B\alpha$ + the numerical factor, the second part of equation [19] less the barometer pressure is the difference in pressure between the two ends, or the centrifugal head.

76 *Static Orifice.* If a static orifice is correct the pressure inside the tip = pressure outside; or atmospheric, at all velocities of air past; then suction k on the manometer at the axis = centrifugal head in the arm. For this case

$$\log \text{ pressure at tip} = 0.0000010479 \frac{W_o N^2 R_t^2}{B\alpha} + \log (B\alpha - K) \ldots\ldots [20]$$

and the number corresponding to the second part of equation [20] less the pressure at the axis $(B\alpha - K)$ is the centrifugal head, K and $B\alpha$ of course being expressed in the same unit of pressure as mercury.

77 *Impact.* Of the runs for impact the first nine were with a fan drawing away the disturbed air. Of these the ones with full gate were affected only very slightly, if at all, by the air motion due to the rotating arm. This point was determined by testing with smoke. The air travel toward the inlet was probably never anywhere more than 300 ft. per min. At the smaller gates there was, however, some motion of the air with the arm so that the runs with full gate deserve more consideration that those at small gate.

TABLE 1 PITOT TUBE CALIBRATION

Test No.	R.P.M.	*A* Manometer Reading, In. Water	*B* Centrifugal Head, Computed	*A*+*B* Pressure in Tip Back of Orifice	*C* Pressure Due to Velocity of Tip in Still Air, In. Water	Pressure in Tip ÷ Pressure Due to Velocity *A*+*B* ÷ *C*	Weight of 1 Cu. Ft. of Air	Humidity Temperatures Dry	Humidity Temperatures Wet	Barometer	Length of Arm Axis to Tip, In.	Date (1908) and Time	Remarks
1	129¾	00	0.489	0.489	0.4888	1.000	0.07247	75°	60°	29.34	41.656	10/15	Impact orifice; ½ gate on fan
2	328½	0	3.112	3.112	3.095	1.003	0.07247	75°	60°	29.34	41.656	a.m.	Impact orifice; full gate
3	388	0	4.4	4.4	4.371	1.004	0.07247	75°	60°	29.34	41.656	a.m.	Impact orifice
4	0 to 480	0	.....	.....	.....	.....	0.07247	75°	60°	29.34	41.656	a.m.	Impact orifice; slow up to full speed. Air motion slow and nearly radial to shaft
5	361	a little +	3.727	3.727	3.709	1.005	0.07192	79°	64°	29.36	41.656	2 p.m.	Impact orifice; air currents change slightly about inlet
6	482	a little −	6.665	6.665	6.612	1.008	0.07192	79°	64°	29.36	41.656	.....	Impact orifice
7	430¾	−⅛ to −$\frac{7}{32}$	5.31	5.18	5.27	0.984	0.07182	79°	64°	29.32	41.656	10/14	Impact orifice, but ½ gate
8	414	0	4.905	4.905	4.870	1.006	0.07182	75°	64°	29.32	41.656	10/14	Impact orifice, but full gate
9	414	−⅛	4.905	4.78	4.870	0.983	0.07182	75°	64°	29.32	41.656	10/14	Impact orifice, but ¼ gate
10	464	−$\frac{5}{16}$	.....	.....	.....	.....	0.06997	92°	78°	29.40	41.62	7/6	$\frac{3}{32}$ in. impact orifice; Std.
11	464	−$\frac{7}{32}$	.....	.....	.....	.....	0.06997	92°	78°	29.40	41.62	p.m.	$\frac{5}{16}$ in. diameter; flat point
12	506.5	−0.23	.....	.....	.....	.....	0.07152	82°	69½°	29.43	41.7	8/15	¼ in. tube open
13	505.5	−0.24	.....	.....	.....	.....	0.07177	78½°	67½°	29.33	41.7	8/18	Std. impact orifice
14	504	−8¼	7.185	−1.065	÷ *B*	−0.148	0.07202	78°	67°	29.41	41.80	8/26	Back end of tip open; ¼ in. tubing tip square
15	506	−9⅜	7.21	−2.165	÷ *B*	−0.305	0.07202	78°	67°	29.41	41.80	3.50 p.m.	Same; but with cord around tube at tip $\frac{11}{32}$ in.

16	508	−8.515	7.28	−1.235	÷ *B*	−0.1695	0.07202	78°	67°	29.41	41.80	.....	5/16 in. tube square as was ¼ in.
17	503.5	−8.44	7.19	−1.25	÷ *B*	−0.174	0.07202	78°	67°	29.41	41.80	.....	¼ in., tube, but beveled 45 deg.
18	507	−0.328	7.4	7.072	7.02	−1.008	0.07202	78°	67°	29.41	41.67	4.00 p.m.	Std. impact; air travel with arm; 245 ft. by anemometer
19	142	−0.562	0.572	0.010	÷ *B*	+0.0175	0.07199	77½°	65½°	29.34	41.625	7/6	Std. static—holes 0.02 in. in diameter
20	193.5	−1.047	1.062	0.015	1.065	+0.014	0.07199	77½°	65½°	29.34	41.625	p.m.	Std. static—holes 0.02 in. in diameter
21	242.2	−1.6875	1.667	−0.020	÷ *B*	−0.012	0.07199	77½°	65½°	29.34	41.625	p.m.	Std. static—holes 0.02 in. in diameter
22	508	−7.32	7.27	−0.05	÷ *B*	−0.0069	0.07208	76°	64¼°	29.29	41.62	8/20	Std. static—holes 0.02 in. in diameter
23	511	−7.34	7.375	+0.035	÷ *B*	+0.0048	0.07208	76°	64¼°	29.29	41.62	5.15 p.m.	Std. static—holes 0.02 in. in diameter
24	512	−7.36	7.4	+0.04	÷ *B*	+0.0054	0.07208	76°	64¼°	29.29	41.62	.....	Std. static—holes 0.02 in. in diameter
25	513	−7.405	7.385	−0.02	÷ *B*	−0.0028	0.07208	76°	64¼°	29.29	41.62	.....	Std. static—holes 0.02 in. in diameter
26	515	−7.46	7.48	+0.02	÷ *B*	+0.0028	0.07208	76°	64¼°	29.29	41.62	.....	Std. static—holes 0.02 in. in diameter
27	505.5	−7.125	7.23	+0.105	÷ *B*	+0.014	0.07216	77°	67°	29.41	41.67	8/26	Std. static—2 holes 0.026 in. in diameter and 2 −0.02 in. and
28	507.5	−7.156	7.28+	+0.124	÷ *B*	+0.017	0.07216	77°	67°	29.41	41.67	2 p.m.	Std. static inner tube—⅛ in. in diameter
29	507.3	−7 3/16	7.28	+0.092	÷ *B*	+0.0126	0.07216	77°	67°	29.41	41.67	.....	Std. static inner tube used in Nos. 27 to 33
30	504.6	−7.125	7.2	+0.075	÷ *B*	+0.010	0.07202	78°	67½°	29.41	41.67	.....	2 holes 0.02 in. in diameter and inner tube ⅛ in. in diameter
31	506.5	−7.156	7.255	+0.099	÷ *B*	+0.0136	0.07202	78°	67½°	29.41	41.67	.....	
32	504	−7.156	7.19	+0.034	÷ *B*	+0.0047	0.07202	78°	67½°	29.41	41.67	.....	
33	506	−7.187	7.25	+0.063	÷ *B*	+0.0087	0.07202	78°	67½°	29.41	41.67	3 p.m.	Same as last; very small leak stopped
34	463	−5.396	6.035	+0.639	÷ *B*	+0.106	0.07050	86°	70°	29.20	41.76	7/10	⅛ in. pipe about 0.28 in. diameter inside with 2 slots opposite 1/16 in. x 2½ in. —static orifice—mean of 4 tests
35	511.2	−7.23	7.33	+1.05	÷ *B*	+0.0143	0.07090	81°	75°	29.21	41.76	8/13, a.m.	Above, with ¼ in. tube inside central
36	501.5	−6.65	7.08	+0.43	÷ *B*	+0.061	0.07057	84½°	75°	29.24	41.76	8/13, 3 p.m.	Above, with inner tube 0.02 in. smaller
37	502.5	−6.4	7.10	+0.70	÷ *B*	+0.099	0.07057	84½°	75°	29.24	41.76	8/13, 3.30 p.m.	Above. No inner tube—same as No. 34

No leak at axis. No inner tube.

78 Another minor source of trouble was the tendency of the stuffing box to heat the air in the tube slightly. This was largely overcome by the very small size and by care to use soft grease freely on the wicking, also by not pressing the outer tube against the wicking harder than necessary. Between runs outside air was allowed to replace the warmed air, but doubtless the results are affected very slightly by the air in the tube being a little warmer than that in the room. The brass tube of the arm always felt as cool after a run as before. Warming of the brass tubes of the stuffing box to as slight an extent as 5 is noticeable to touch at these temperatures.

79 An increase of 10 deg. in temperature in run No. 8 without change of moisture in the air in arm, changes the computed centrifugal head from 4.905 in. at room temperature to 4.825 in.

80 Of the first nine runs only No. 6 at the highest speed shows a negative pressure in the water manometer used, when the fan was fully open. With the smaller gates the readings were slightly negative. The mean of these runs is 0.9991 for the pressure in the tip divided by the pressure due to velocity. However, considerably more weight must be given the runs with full gate. If the runs with the smaller gates be left out then the pressure in the tip exceeds that due to velocity by an amount that would have been noticed and measured, for the water levels in the manometer used were steady and the conditions good. It is believed that the slight rise in temperature of the air in the tubes accounts for the observed pressure being slightly less than the computed.

81 Runs Nos. 10 to 18 inclusive were all made without the fan to draw the disturbed air away. Nos. 10 and 11 show a comparison of a flat impact point $\frac{5}{16}$ in. in diameter with a hole in the center about $\frac{1}{16}$ in. in diameter with the same tip used in the first runs, marked Std. Nos. 12 and 13 compare a $\frac{1}{4}$-in. open brass tube with end filled square with Std.

82 Nos. 14 to 17 were with the back end of the tip open (see *B* in Fig. 2), using four forms of opening. No. 18 is given to show about what air motion the arm set up for these runs. These readings are given because nearly every one interested in the pitot tube sooner or later wants to know what this suction is. The suction evidently depends for one thing on the shape of the orifice and the tip.

83 In No. 18 the pressure due to velocity may be found from the velocity of tip less the air motion

$$\text{Tip velocity} = \frac{41.67 \text{ in.} \times \pi\, 2 \times 507}{12} = 11080$$

$$\text{Air motion in path and moving with tip} = 245$$

$$\text{Velocity of impingement on tip} = 10835$$

From

$$\text{velocity, ft.} = 1097 \sqrt{\frac{\text{velocity pressure}}{\text{weight 1 cu. ft. air}}}$$

$$\left(\frac{10835}{4085}\right)^2 = 7.02 \text{ in.}$$

which is the velocity pressure due to the relative velocity of the tip to the air in inches of water. The centrifugal heads in column *B* were computed by logarithms by the method in Par. 74 except that the final head in inches of mercury was multiplied by 13.61 by slide rule.

84 For computation of centrifugal head for the static runs pressure at the axis is $B\alpha - k$, if the pressure in the tip is atmospheric, and the mean density in the arm is slightly less than for the impact runs, since for the impact runs pressure at the axis is very nearly atmospheric.

85 It can be shown that the pressure due to impact on the tip, Fig. 2, would exactly equal the centrifugal head if the fluid in the arm were of uniform density.

86 *Static.* Static runs Nos. 19 to 33 are with good forms of static orifice. It is thought that the two sets of runs 22 to 26 and 32 and 33 deserve to carry greater weight than the others. However, the mean of all for pressure in tip divided by the centrifugal head is 0.00625. It is evident by comparing the first part of these runs with the last that the holes are small enough so that the static does not depend on the inner tube in any way.

87 The static orifices are simply holes 0.02 in. in diameter drilled through the tubing which was about 0.03 in. thick. The holes were clean and the outer edges square, the little burr being carefully filed off.

88 Runs Nos. 34 to 37 are with slotted static orifice. The outer tube was about 0.4 in. in diameter, its inside diameter being about 0.28 in. With no inner tube the results are poor even with this thick tube. With an inner tube central and fitting the outer within about 0.015 in. the result was fairly good, but when the inner tube was made 0.02 in. smaller in diameter the result was poor again. It is very interesting to know that when fine mesh wire cloth was wrapped around the tube outside the slots the results are good and agree with those from the small drilled holes. Just how accurate the gauze static would be was not found.

89 By comparison with standard static in air flowing in a pipe the results are found to be considerably worse with a thin outer tube than with a thick one using the same inner tube. An instrument with $2\frac{1}{4}$-in. slots $\frac{1}{16}$ in. wide in $\frac{3}{8}$ in. tubing $\frac{1}{32}$ in. thick and with $\frac{7}{32}$ in. inner tube gave static pressure readings 14 per cent of the velocity pressure too high.

90 By comparison in this way it was also found that holes $\frac{1}{16}$ in. in diameter in the $\frac{1}{4}$ in. regular tip which is $\frac{1}{32}$ in. thick give poor results.

DISCUSSION

W. H. Carrier stated that the same kind of pitot tube as that used by Mr. Treat had been used for a long time at the works of the Buffalo Forge Company, however, instead of using two tubes side by side, connecting to the static member and to the total pressure member, one tube is placed inside of the other, so that the pitot tube can be more easily inserted in the pipe. In this construction there is only one pipe to be handled, and it requires only about one-half as long to construct as Mr. Treat's.

There is one point worthy of particular attention as the feature probably responsible for a very large amount of inaccuracies in fan tests and misleading statements as to efficiencies, such as claims of efficiencies of 83 per cent or even as high as 90 per cent, which cannot be attained in ordinary fans for low-pressure work. The actual efficiency of those of the very best design working under the best conditions is 70 to 75 per cent, while fans of the multivane types of ordinary commercial design have efficiencies varying from 45 to 60 per cent only. The reason why unusual efficiencies are shown in some tests is in all probability because of a slight leak in the static member, which usually happens unless precautions are taken. This leak reduces the static pressure and a further reduction may be produced by some of the minute openings being clogged by dirt. Even a small reduction, say 5 per cent, will produce an apparent error in the reading of the velocity pressure of about 15 per cent, which will result in the amount of air calculated as delivered being 7 or 8 per cent larger than is actually the case. The error due to these causes may often be as large as 20 to 30 per cent. The proper method is to test the connections to both the static and total members as to tightness by blowing into the connecting tube, depressing the water column, and then holding it, with the thumb over the opening, to see if any drop occurs in the water column.

Sanford A. Moss. Experience with velocities of the order of magnitude of those used by Mr. Treat as well as very much higher velocities has shown that Mr. Treat is correct when he states that for any ordinary change in an impact tube no inaccuracy results but that a static orifice which reads correctly is not easily obtained. I have had most successful results in measurement of static pressure by means of a hole in the pipe wall, succeeded by some length of straight pipe and preceded

by a considerable length of straight pipe. With the static pressure obtained in such a way the pitot tube becomes merely an impact tube.

Some years ago I used an arrangement such as is shown in Mr. Treat's Fig. 3 with a jet discharging into the atmosphere. Mr. Treat, however, seems to use a jet discharging from a long straight pipe of constant diameter. In such a case the velocities at the point of discharge are variable and gradually decrease from the maximum value at the center to a small value close to the edge. As is well known, the curve giving these velocities for various diameters is an ellipse. I used an orifice consisting of a gradually rounded approach succeeded by a parallel portion about one diameter in length and preceded by a long straight pipe with a diameter about three times that of the parallel portion at the discharge of the orifice. It can be shown that such an orifice gives a jet with practically uniform velocity and that the pressure in the jet must be atmospheric.

I inserted a pitot tube similar to that used by Mr. Treat in such a high velocity jet and found that for high velocities static pressure was very slightly less than atmospheric. For a velocity of about 400 ft. per sec. the static pressure was 0.6 in. of water below atmosphere. For velocity up to about 80 ft. per sec. the static pressure was practically atmospheric. Other forms of pitot tube in such a jet showed static readings with much greater errors.

Mr. Treat's suggestion regarding an inclined gasolene tube for pressures of less than 1 in. of water is very interesting. I would like to inquire how he secures an exact value of the zero angle, how he secures the density of the gasolene and how he avoids change of density by evaporation of the more volatile part of the gasolene from time to time.

The variation of pressure with square of velocity and power with cube of velocity for a fan discharging against constant resistances mentioned by Mr. Treat was considered theoretically by Professor Rateau many years ago and is undoubtedly a very exact relation so long as the pressures are so low that the change of density is not appreciable.

All of Mr. Treat's methods seem very carefully and accurately laid out and it is pleasant to know that the measurement of quantity in his fan blower work is being carried out with precision.

J. H. Browne said he would like to see some method developed by which the pitot tube could be used when the fan is used to produce either forced draft under a boiler or induced draft over a boiler. This would show the exact conditions of operation better than the ordinary water tube or inclined gage because in the latter case the thickness of the fire affects the results.

C. G. de Laval (written). On using a pitot tube for measurement of air one must not overlook the question of moisture in the air as this affects the air density; also, when water is present there will be a wide variation in the velocity, so that to be fully accurate relative humidity should be taken into account. In the usual cases the velocity is not constant in the cross-section and particular care has to be exercised to read the pitot tube carefully; wrong records may be due to fluctuations, and coefficients of corrections have to be used in order to get the correct measurement. The paper aims to show a tube that would give accurate results in a simple way; the best and most reliable is one that will combine the impact and static in one, and in such a way that the dynamic and aspiration effects neutralize each other so that the real pressure is obtained.

For measuring air velocity by the formula $v = \sqrt{2gh}$, the density of air and its velocity head, temperature, atmospheric pressure and moisture must be taken into account. The density

$$D = \frac{0.0807\ (p - 0.379\ hv)}{(1 + zt)29.921}$$

where

0.0807 = weight of 1 cu. ft. dry air at 32 deg. fahr. and atmospheric pressure of 29.921 in.

p = atmospheric pressure

h = relative humidity

v = vapor pressure

$1 + zt$ = ratio of volumes of dry air at 32 deg. fahr. and at $(32 + t)$ deg. fahr.

t = temperature of dry air above 32 deg. fahr.

0.379 = per cent of vapor in 1 lb. saturated air at 32 deg.

It is believed that a single tube at center will correctly register the velocity with a correction of 4 to 5 per cent for the average over the entire pipe or outlet and that it will be necessary to correct further for moisture in the air; it will be also found that

the velocity of the air materially affects the correctness of readings by pitot tubes. For low velocities, say below 5000 ft. per min., a special form of gages will have to be used in order to be reliable and to register accurately the velocity heads, while for higher velocities the instruments of the pitot tubes are fairly reliable as to readings.

The electric meter is probably the most accurate, it measures the energy necessary to cause change in temperature, and this is proportional to the weight of air flowing.

$$Q = \frac{E}{t \times s}$$

E = energy to raise temperature
Q = weight of air through t deg.
s = specific heat of unit of weight at constant pressure.

The electric meter is independent of variation in velocity of the air at the different parts of pipe. Measurements used are all electric and can be read to close observation and are not affected as in the pitot tube and venturi meter by eddies and pulsations. It is to be noted that neither pitot nor venturi meters will or can lend themselves to continuous graphical results due to the small power available for recording, while the electric meter of Professor Thomas can give a continuous reading through an ordinary wattmeter, and which is still better, the readings can all be obtained without any calculation, as it necessitates only the electrical energy transformed into standard cubic feet, and its accuracy depends only on its ability of measuring electrical energy.

In closing, Prof. C. C. Thomas, who presented the paper, stated, in reply to a question from Mr. Moss, that in regard to the use of light fluids such as gasolene for measuring the difference between static and velocity pressures, there was some difficulty at times due to the varying vapor pressures in the gages. Almost all such gages are closed at one end, and there is a different pressure above the fluid for each temperature, so that in summer it is difficult to get the same results as in winter.

The range of the pitot tube stations is determined by the smallest and the greatest heights of water columns which it is found practicable to read accurately, and to obtain. About 4 in. is the lower limit in the station with which the speaker is familiar, and 28 in. the higher. This gives the tube a measuring range of the square root of 7 or about as 1 is to 2½.

THE AUTHOR. Mr. Carrier brings out a point worthy of particular attention when he places the blame for some of the discordant results of tests on leakage. I would not limit to leakage on the static side only, but anywhere in the connections. It is important that these easily cared for precautions be taken in any testing work and no less so in pitot tube air measurement.

In trying for leaks, pressure is best applied at the pitot tube orifices so that all the piping will be just as it is to be for the test. This method is noted in Par. 24 together with a caution against stopping the connections and drawing over liquid into the rubber tubing. These points should be carefully looked into before every test and it has been the regular practice in work under the writer's charge.

In regard to the detail of arm carrying the pitot tube the only objection to one tube inside the other is the smallness of the inner or the largeness of the outer one. In the first case it is difficult to clean the inner tube of dust or of liquid if it comes over from the manometer. In the second case the size of arm is objectionable. A better way of making the arm is to extend it through the measuring pipe at both sides. What slight obstruction there is to smooth flow is then constant for all positions of the pitot tube across the pipe.

I cannot subscribe to all of Mr. Carrier's statements as to fan efficiencies.

I am grateful to Mr. Moss for putting into the discussion some of the results of his experience in this field.

In regard to setting inclined manometers at zero angle much the best way found is to fill the tubes nearly full of gasolene, leaving only good length air bubbles, corking up the open ends and using the tubes themselves as a level. If the tubes are good this affords means for very accurate setting. After setting the gage the excess gasolene is run out leaving them about half full ready for use.

Our manometers had graduations for different positions to give 2, 5, 10, 20, and some of them 40 slant to one vertical and the zero angle on the circumference of the front plate and the reference line on the back plate. With the back plate set with its reference line opposite the zero line on the front plate when the tubes show level, any slant may be had by swinging to it.

The graduations need to be fine and accurately placed and at a good radius from the pivot, which should be well made so that

there is no lost motion. These precautions are essential for good readings at the small angles 20:1 and 40:1.

The specific gravity of gasolene for each new canful, when filled to a hook gage mark in the neck of the can is usually determined by the weighing method. As two manometers are almost always used, it is convenient to check the specific gravity from time to time using water in one manometer and gasolene in the other and applying the same pressure to one leg of each, the height of water being divided by the height of gasolene. A can of gasolene if kept fairly well stopped does not change much in specific gravity, nor does that left in the manometer change fast. This point of density of gasolene must be watched, however, and the second method affords a convenient way.

As to gasolene manometers there is some apprehension in regard to gasolene vapor in the connections affecting the readings. The apprehension is found to be groundless when it is tried out. Even when the manometer is set to the smaller angles the tube and connections can be swung from several feet above to as many below without any noticeable change in the manometer reading.

In the use of manometers in this pitot tube work, whether water or gasolene be used, any change in the volume in the connections from evaporation of the liquid is freely vented through the tube orifices or at the open leg of manometer when both legs are not connected, and therefore no error can result from vapor pressure.

Fig. 14 shows an inclining manometer as described. The upper instrument is an inclining averaging manometer by means of which the mean reading of ten pitot tubes can be obtained. It is connected to give average velocity pressure across the pipe. The left leg of the manometer is connected to a static of one of the pitot tubes. This arrangement for static is usually correct, but when the flow in the pipe is disturbed the static is not uniform and constant across the measuring pipe, and static and total or impact must be taken separately. For instance, the discharge from a fan with outlet open cannot usually be measured in this way, using only one static.

In Figs. 15 and 16 the areas and sections of the several tubes are all made the same. If AA is the level of the liquid when all tubes are open and if the liquid is depressed in the right-hand tube amounts a, b and c, as the areas are the same the displaced

liquid must rise in the left tube an amount

$$a+b+c=o$$

Add o to a and we have A, $b+o=B$, $c+o=C$

$$a+o \quad + \quad b+o \quad + \quad c+o \quad = \quad 4 \quad o$$

or

$$A \quad + \quad B \quad + \quad C \quad = \quad 4 \quad o$$

and the mean reading

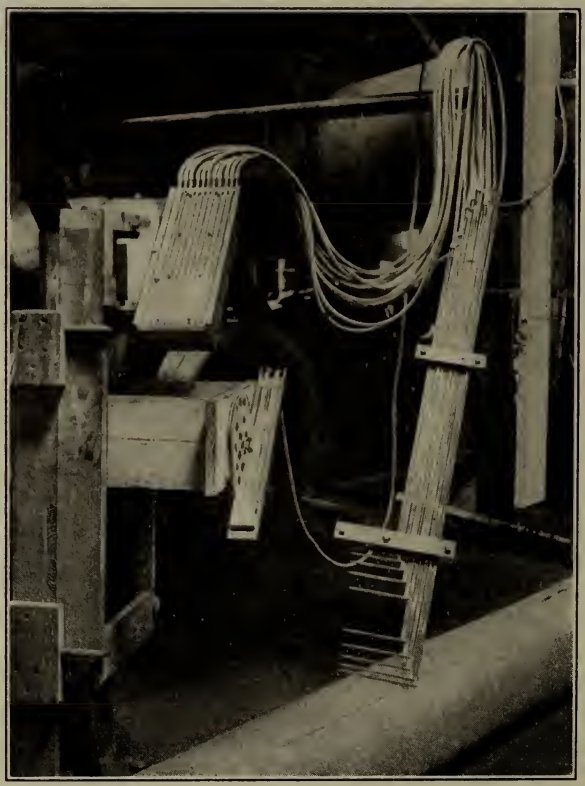

FIG. 14 PITOT TUBES READY FOR MEASURING ACROSS PIPE WITH INCLINING GASOLENE MANOMETERS

$$\frac{A+B+C}{3}=\frac{4\,o}{3}$$

For any number of manometer tubes n we have for the mean reading

$$\frac{A+B+\text{etc.}}{n-1}=\frac{n}{n-1}o$$

In use the distance o requires to be taken and multiplied by $\frac{n}{n-1}$.

So far as known this is a new form of manometer. It is interesting to watch the slight fluctuations in some cases of pressure across a measuring pipe. Plate glass was used over grooves milled in a brass body for this instrument.

The inclining manometer is accurate for small heads if the tubes are parallel and straight. The accuracy was tried out by simultaneous measurement of the pressure in connecting tubing by such a manometer at a 40 to 1 slant, by a hook gage, by a good manometer and by a gasometer arrangement suspended inside a chemical balance. The measures checked very closely or to about 0.001 in. of water.

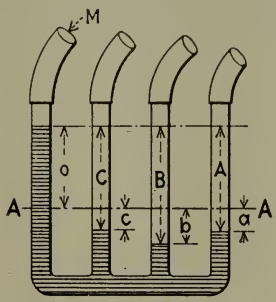

FIG. 15

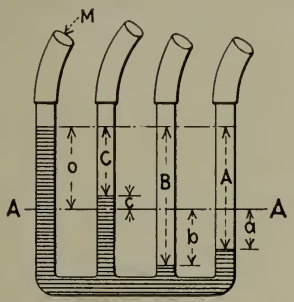

FIG. 16

EQUAL AREAS AND SECTIONS IN PITOT TUBES FOR USE WITH INCLINING MANOMETER

Mr. de Laval confers a favor by calling attention to the matter of determining the density of air under measurement giving a formula for density. The tables published by the Navy Bureau of Construction and Repair show the weight of air for different temperatures, barometer readings and humidity up to 100 deg. and are a time saver within their range. Formula [15] applies to any gas by using the density of the gas measured. (See also Pars. 44 and 45.)

As to measuring the velocity of air flow in a pipe with the pitot tube at the center, the correction for average velocity should usually be determined by trial and especially if the pipe is short or if flow is in any way irregular. If gates are used on the measuring pipe it is necessary or the error may be considerable.

As to the velocities of air affecting the correctness of the pitot tube readings my evidence is that these readings are correct at least to the limits stated and up to the speeds experimented with. That is, both static and impact orifices if they are as in Fig. 1 give true static and true impact pressures. The method of calibration is simple and direct especially for static orifice.

Possibly Mr. de Laval's experience has been with some of the forms having a faulty static orifice with which calibration was required to determine a constant in the relation $v = \sqrt{2gh}$. The form shown in Fig. 1 is simple and easy to make, the tip consisting of a $\frac{1}{8}$-in. tube inside a $\frac{1}{4}$-in. diameter tube with walls about $^1/_{32}$ in. thick. Both static and impact are correct if made as shown. Accurate pressure measuring instruments are available and must be used and the ordinary precautions taken against leaks, etc., as noted through the paper and in Pars. 18 to 25.

No. 1374

THE V-NOTCH WEIR METHOD OF MEASUREMENT

By D. Robert Yarnall, Philadelphia, Pa.

Member of the Society

Attempts to measure accurately and to record automatically the flow of water through pipes or channels have been made from time to time with varying results as to accuracy and convenience. This paper will attempt to show results obtained by the simple and practical V-notch method of measurement, which though not new, seems now to be meeting with marked success, both abroad and in America, in power plant measuring problems.

2 Two tests of the accuracy of this method of measurement have recently been made in America and the data collected are here presented to show the degree of accuracy attainable.

3 The instrument employed in the tests is the Lea V-notch recorder of which a description follows:

4 This apparatus depends on the laws governing the flow of water through weirs, which laws have been found to work with extraordinary accuracy. It is only comparatively recently, however, that they have been turned to account in measuring widely varying rates of flow, both hot and cold, met with in power plants.

5 The most common form of weirs is the plain horizontal sill and the sharp edge rectangular notch. The former of these has the full width of the stream or channel and is without end contraction, the formula used in connection with it being that of Francis

$$\text{cubic feet per second} = 3.33\ LH\sqrt{H}$$

where

L = the width of the weir in feet.

H = the head of water passing over it in feet.

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6 The second weir is a sharp edge rectangular notch of less width than the channel or stream and therefore has end contraction. The formula used in this is

$$\text{cubic feet per second} = 3.33\, LH\, \sqrt{H \times C}$$

where

L and H are the same as before.

C = a constant which depends on the ratio of the width of the weir to the width of the channel.

7 Although these two formulae give a fair degree of accuracy there is another case in which the accuracy is much greater and approaches very nearly 100 per cent. This is the

TABLE 1 FLOW THROUGH 90-DEG. V-NOTCH WEIRS

Depth in Notch, In.	Flow per Hour, Lb.	Depth in Notch, In.	Flow per Hour, Lb.
1	1,140	9	277,960
2	6,480	10	361,740
3	17,830	11	459,030
4	36,610	12	568,720
5	63,940	13	694,710
6	100,860	14	836,110
7	148,290	15	993,510
8	207,060		

sharp edge V-notch of Prof. James Thomson, the formula of which is

$$\text{cubic feet per minute} = 0.305\, H^2 \sqrt{H}$$

where

H = the height of the notch in inches, the angle of the notch being 90 deg.

8 Table 1 gives the flow of water through 90-deg. V-notches for each inch of head, up to and including 15 in., calculated by Thomson's formula.

9 The section of the flow through the V-notch is at all times the same shape, though the area may vary, and this constancy of form tends to simplify the formula and make it accurate. This form of notch is also especially adapted for measuring smaller quantities of water than the rectangular weirs, as shown by the curves in Fig. 1 which give a comparison of the accuracy of the different types of weirs at different rates of flow. Another property of the notch is that its angle may be less than 90 deg.

without impairing its efficiency, which enables it to be used to measure very small quantities of water.

10 It was for these reasons, as well as for its accuracy, that the V-notch was adopted for use in connection with the Lea recorder. A sectional view of the apparatus is shown in Fig. 2. It will be seen that a float spindle, rigidly attached to a float

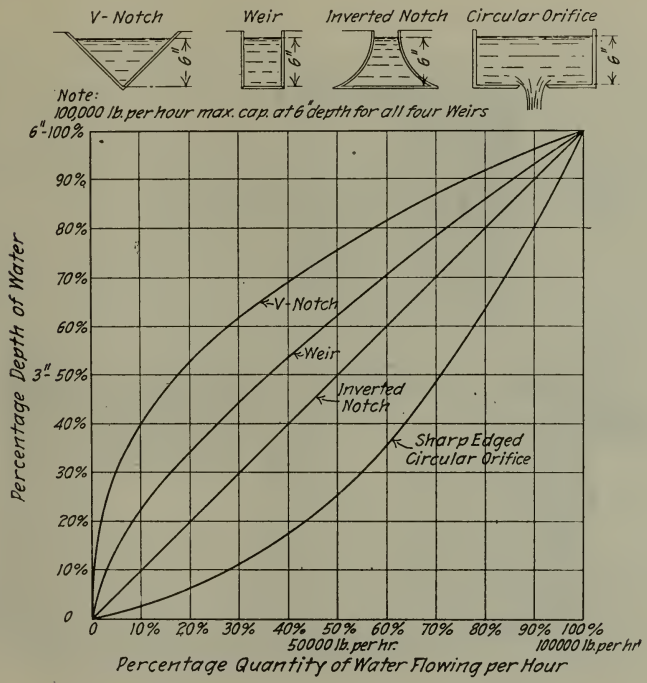


Fig. 1 Curve showing Comparative Accuracy of Different Types of Weirs under Different Rates of Flow

which rides on the surface of the water flowing over the V-notch, passes up through the bottom of the instrument case. A rack on this spindle gears into a small pinion upon the axis of a drum revolving between centers.

11 Upon the body of the drum is a screw thread, Fig. 3, the contour of which is the "curve of flow" for the V-notch in connection with which the recorder is used, and as the flow through the notch increases rapidly with the depth, the pitch of the screw increases in the same proportion. Above this drum is a horizontal slider bar, supported on small pivoted rollers and carrying an

arm, at the upper end of which is a pen or pencil point in contact with a paper chart upon a clock driven recording drum, which revolves once in 24 hours (Fig. 4). As the float rises or falls the drum spiral is rotated, and its motion is imparted to the slider bar and pen arm by means of a saddle arm projecting from the latter and engaging at its lower end with the screw

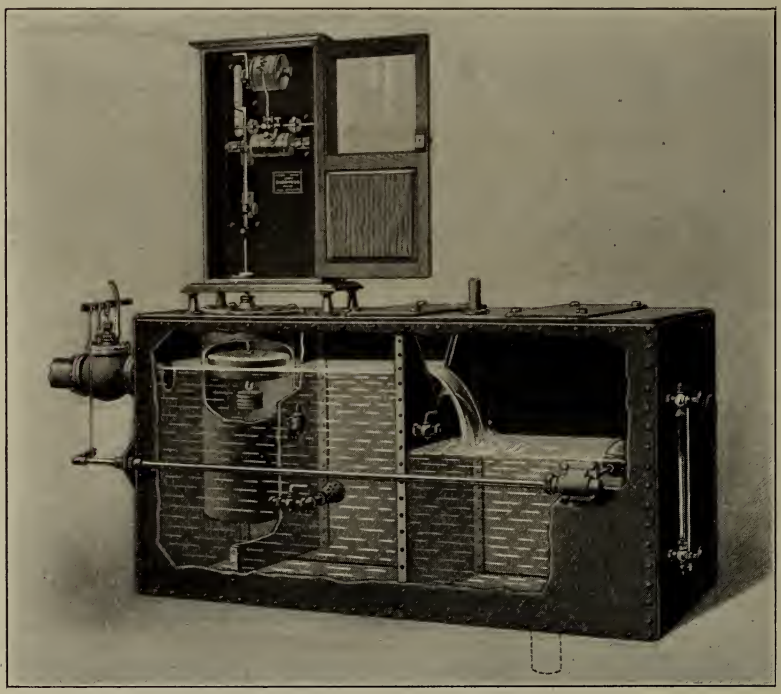


FIG. 2 SECTIONAL VIEW OF APPARATUS USED IN TESTS

thread. On the body of the spiral drum is a graduated scale adjacent to the screw thread and computed in pounds per hour. It is thus seen that the drum serves the double purpose of rectifying the motion of the pen so that it moves equal distances for equal increments in the rate of flow; and of providing a magnified scale for making an accurate observation of the rate of flow at any moment.

12 The apparatus has the following advantages:

a It is simple and hence dependable.

b Its operation is continuous.

c There are no moving parts in contact with the flow except the float, which is in a chamber by itself and not in the path of the flow.

d Collections of sediment and deposits of scale do not affect the accuracy.

e Owing to change of density of the liquid being measured due to changes in temperature, the float which actuates the recorder is immersed more or less, de-

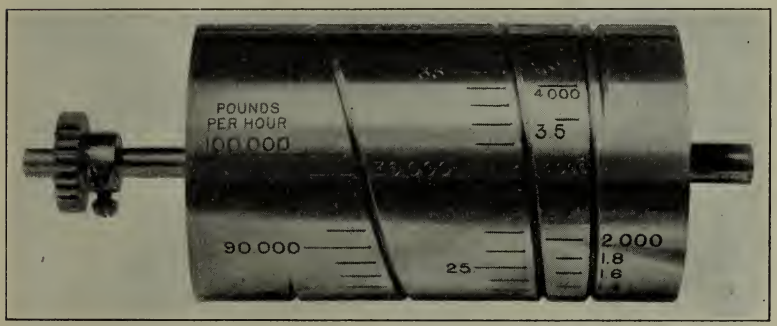


Fig. 3 Detail of Recorder Drum showing Graduated Curve of Flow

pending upon this change of density, and the recorder and the float are so constructed that they compensate automatically for these changes in temperature over a wide range.

f By making the tank of ample proportions a considerable interval of time takes place between changes in the head on the weir, so that the recording apparatus has a chance to adjust itself to changes in the rate of flow, thus giving a chart very easy to measure with the planimeter.

g It is always placed on suction side of pump, hence it is available at all times for inspection, cleaning and repairs.

13 Like the venturi meter and the pitot tube, the V-notch recorder is a continuous flow meter, but unlike these it is not dependent for its accuracy on velocity changes, and hence its record is not subject to the fluctuations due to the sudden changes

of velocity head when these meters are in service, measuring rapidly changing rates of flow.

14 Perhaps the greatest advantage of this method of measurement is found in the way by which it may be checked independently of the recording mechanism. The operator may construct alongside of his recorder an independent "hook" gage or use the graduated vertical scale and pointer on the extended float

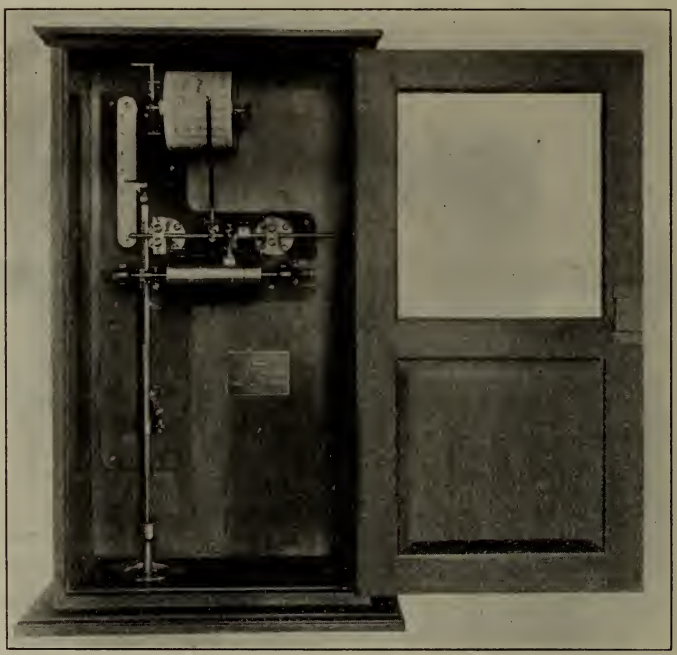


FIG. 4 DETAIL OF RECORDER

rod in the instrument, and can observe from time to time the number of inches head on the weir. By substituting these values in Thomson's formula in Par. 7, the rates of flow as shown on the recorder chart may be checked quickly and accurately. Thus the operator need not depend upon the recorded result until he is sure that it agrees with the figured result from his hook gage readings. The author believes, therefore, that the V-notch with its tank is the most satisfactory testing set that could be devised for measuring large and small rates of flow, hot and cold, as well as fluctuating or uniform conditions of flow.

15 For results the following reports of tests are given, and as clearly as possible the actual test conditions which prevailed explained.

16 With a view to checking the accuracy of this apparatus and the V-notch for the measurement of boiler feedwater a series of tests were made at the Dartmouth College power plant, at both winter load and summer load. These tests were conducted

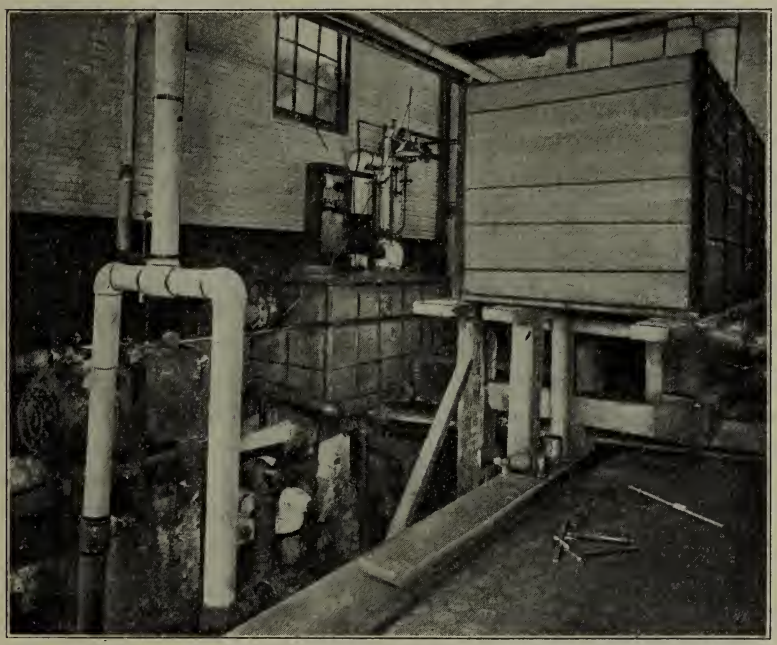


FIG. 5 ARRANGEMENT OF TESTING APPARATUS FOR THE DARTMOUTH COLLEGE TEST

by the engineers for the college, Messrs. R. D. Kimball Company, through their representative, J. H. Browne, assisted by the writer.

DARTMOUTH COLLEGE WINTER TEST

17 Fig. 5 is from a photograph of the testing apparatus and meter. On the right-hand side is a large wooden tank divided into two parts by a vertical partition. The size of this tank was 54⅝ in. deep by 54¼ in. wide by one section $59^{1}/_{16}$ in. long and the other section 59⅛ in. long, inside measurements. To the

left is the cast-iron V-notch tank and mounted above is the recording apparatus. The inlet of the meter is controlled by a 6-in. balanced float valve operated by a float riding on the surface of the storage section of the notch tank below the weir, so as to prevent flooding in the event of the feed pump slowing down or stopping (Fig. 2).

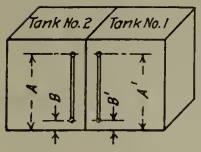
18 The return water from all of the buildings of Dartmouth College is collected in the receiver shown at the left of the meter tank. The make-up water also enters this receiver, which is under about 5 lb. back pressure. The 4-in. line discharges from this receiver into either one of the sections of the wooden measuring tanks, the water flowing by gravity from this tank through the meter to the feed pump and is thence pumped into the boilers in the adjoining boiler plant.

19 The two sections of the wooden tank were provided with independent vertical glass columns ¾ in. in diameter. Just back of and touching each glass column was provided a scale graduated in sixteenths of an inch. The two tanks were carefully calibrated by actually weighing the value of 1 in. in height of the water in the tank at a given temperature, this value being independently computed as noted in the tables. It was found during the test that the meniscus of the column at the upper and lower reading could be read with great accuracy because of the careful construction of the column and the graduation of the scale.

20 It is to be noted that this test was made at the normal running load and took care of all of the natural fluctuations in the returns from the heating system. The test was conducted when the outside temperature was only slightly above zero most of the time, hence the heating system was being taxed to its utmost.

21 The test was run under exact working conditions of the plant at heavy load, which fluctuated badly (Fig. 6A). These fluctuations in the load were caused by poor control of the return water from the new gymnasium which was at a lower elevation than the power plant and the accumulated returns from the heating system were intermittently pumped up into the power plant by a float controlled motor driven pump. Only very small water storage was provided in the receiver, hence the fluctuating boiler feed conditions shown; under these conditions the results of the test are remarkably accurate.

TABLE 2 TEST OF V-NOTCH RECORDER AT DARTMOUTH COLLEGE, MARCH 23, 1912



$A - B = C$

$A' - B' = C'$ Correct Quantity $= \Sigma (C + C')$

METER CAPACITY, 50,000 LB. PER HR.; 6-IN. RECORDER; 90° V-NOTCH 6 IN. DEEP

	TANK NO. 1			TANK NO. 2			
Time	A	B	$A-B$	A'	B'	$A'-B'$	Temperature, Deg. Fahr.
9.00	37 3/16	7 5/8	29 9/16	42 5/8	5	37 5/8	164
9.22	42 1/2	3 5/8	38 7/8	34 1/2	3	31 1/2	
9.41	32 7/8	3 1/8	29 3/4	39 7/8	2 3/4	37 1/8	180
9.50	41 1/8	3 1/4	37 7/8	42 3/4	8 3/4	34	
10.15	43 7/8	7 1/8	36 3/4	43 1/8	3 3/8	39 3/4	180
10.40	42	2 3/4	39 1/4	39 7/8	4 1/2	35 3/8	
11.06	40 1/4	2	38 1/4	42 3/4	3 1/2	39 1/4	170
11.40	44	3 7/8	40 1/8	44	7 1/4	36 3/4	
12.00	43 1/2	3	40 1/2	43 1/4	3	40 1/4	160
12.27	44	5 3/8	38 5/8	44 5/8	2 1/2	42 1/8	
1.00	44	2	42	44	1 3/4	43 1/4	195
1.25	44 3/8	2 1/4	42 1/8	44 1/4	2 1/8	42 1/8	
1.50	46 1/4	4 3/8	41 7/8	44 1/2	2 7/8	41 5/8	195
2.15	43	2 1/4	40 3/4	44	5/8	43 3/4	
2.52	39 3/8	3 3/8	38	43 7/8	2 3/4	41 1/8	194
3.25	44	1 1/8	43 7/8	41 1/8	3 1/2	37 5/8	190
4.00		Total.... 615 3/16 in. or 615.18 in.			Total.. 621 7/8 in. or 621.87 in.		

Tank calibration by actual weight at 175 deg. fahr.:

In tank No. 2, 1 in. = 117.0 lb.

In tank No. 1, 1 in. = 115.2 lb.

615.16 × 117.0 =	71976	
621.87 × 115.2 =	71640	
Total.....	143616	
Error due to tank leakage	225	143841

From planimeter measurement of chart (see Fig. 6A)

Quantity, lb. = Area × Constant

4.04 × 36312 = 146700.48

Correction for zero adjustment of recorder.....	1162	145538
Recorder error (high) =		1697
Recorder error, per cent =		1.18

DARTMOUTH COLLEGE SUMMER TEST

22 The second test was made in the summer to check accurately the claim that the meter is as accurate at low rates of flow as at high. Table 3 substantiates this claim as the accuracy was even greater (0.55 per cent) than that at the winter

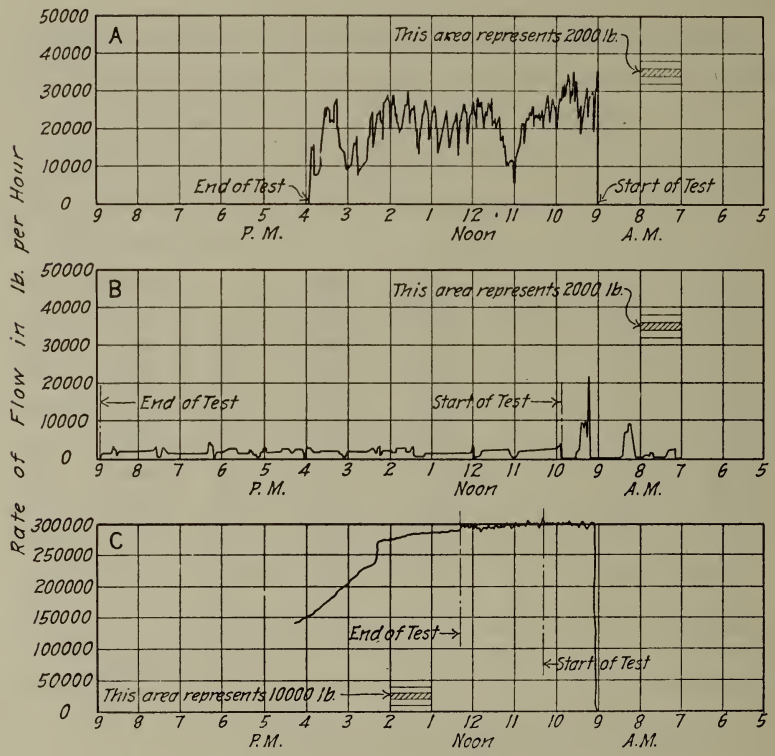


FIG. 6 RECORDING CHARTS

load. This due to the more even rate of flow (see low reading) which in turn is due to the gymnasium heating system not being in service.

TESTS ON V-NOTCH METER IN PHILADELPHIA, AUGUST 9, 1912

23 Figs. 7 and 8 show the arrangements for another test conducted recently in exactly the same way as the tests at Dartmouth

College except that the water was measured on the discharge side of weir instead of the intake side. In this test, however, the water being measured was at 73 deg. while the Dartmouth tests were run at an average of 190 deg. fahr.

24 The V-notch meter was mounted for convenience on a platform car on the coal siding shown in Fig. 8. The car during the test was rigidly jacked to prevent vibration.

TABLE 3 TEST OF V-NOTCH RECORDER AT DARTMOUTH COLLEGE, AUGUST 2, 1912

Tank No. 1				Tank No. 2			
Time	Full, In.	Empty, In.	Weight	Time	Full, In.	Empty, In.	Weight
10.52	32.0	....	.....	9.43	44.81	6.50	4495.3
11.47	....	1.68	3502.3	11.41	32.44	7.81	2890.0
1.03	40.0	0.875	4520.8	2.03	33.81	1.75	3761.9
2.55	34.5	2.00	3655.4	3.54	41.50	1.63	4680.1
4.55	41.25	9.125	3712.0	6.00	45.31	1.19	5177.0
7.15	40.25	7.06	3880.8	8.00	41.38	29.87	1349.4
		Total.....	19271.3				22353.7
							19271.3
						Total.....	41625.0

Chart area = 0.570
Constant = 36312. lb. per sq. in.
Product = 20697.84
As 90 V-notch is used multiply by 2 as drum is cut for $\frac{1}{2}$ 90..... 41395.7
Error..... 2.293
Result..... 0.55%

Below the notch tank are shown two large cast-iron tanks, made of sectional plates 10 ft. 8 in. long by 8 ft. wide by 6 ft. 2 in. high, inside dimensions. The water from the meter could be discharged into either one of these measuring tanks by throwing over a tipple without interrupting the flow. These measuring tanks were provided with carefully constructed glass water columns with accurately graduated scales back of the glass. Baffle plates were constructed within the measuring tanks to prevent wave motion and resulting fluctuations in the water columns. Below the measuring tanks is shown the large storage supply tank 22 ft. long by 8 ft. wide by 6 ft. 2 in.

high, also built of cast-iron sectional plates, into which the upper tanks were in turn drained. The measuring tanks were very accurately calibrated by weighing on a corrected scale a unit charge of 100 lb., marking the scale through a certain range near the top and bottom of the tank for each unit charge of 100 lb.

25 The test as will be observed was run at approximately

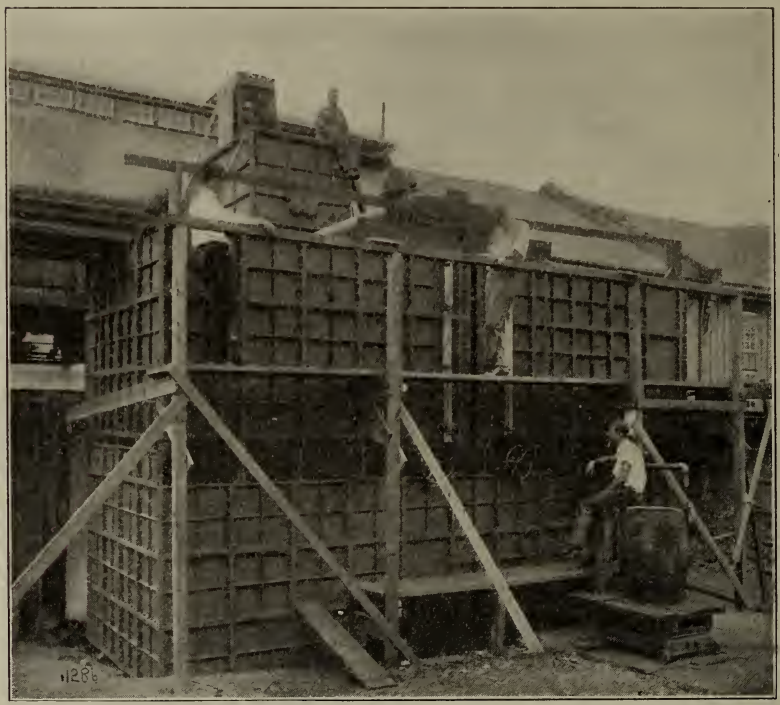


FIG. 7 ARRANGEMENT OF TESTING APPARATUS FOR PHILADELPHIA TEST

the rate of 300,000 lb. per hr. This was the maximum capacity of the recorder under test. A chart record was obtained during the test which was measured carefully in order to get the quantity for comparison with the quantity measured in the tanks below the weir tank. The recorder had a drum with a curve of flow for use at a temperature of 212 deg. As the water during the test was 73 deg. it was necessary to make a correction for this large difference in temperature. This correction has been applied in the data of the test as will be noted. Correction is also made for inaccuracy of scale as found by calibration.

Duration of test = 1: 57.86 = 1.9643 hour.
By scaling chart = 1.09375 in. = 1.9643 hour.
Hence recorder clock driving chart is correct.

Then 1 in. of chart $= \frac{1.9643}{1.09375} = 1.796$ hours.

By scaling chart (vertically) 2.5 in. = 300,000 lb. per hr.
Therefore 1 in. = 120,000 lb. per hr.
Therefore 1.796 × 120,000 = 215520 lb. = 1 sq. in. in Fig. 6C.

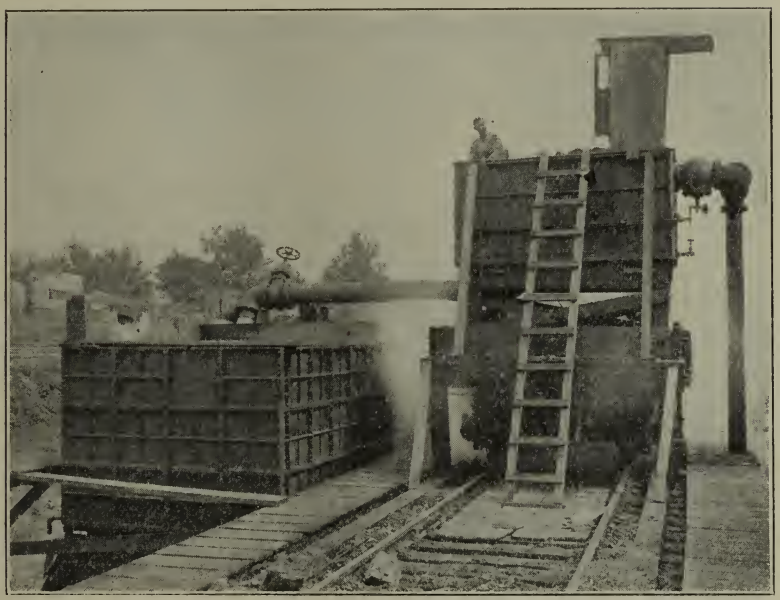


FIG. 8 ARRANGEMENT OF TESTING APPARATUS FOR PHILADELPHIA TEST

26 Integrating chart area under curve = 2.70 sq. in.
Hence 215520 × 2.7 = 581,904.0 lb. = recorder record.

27 Error in scales: each 400 lb. water as graduated at gage glass on tank is actually 400.75 lb. by correct weight, each pound of water passing through the meter cold (about 68 deg.) will increase in volume to such an extent when hot as to require the addition of 3½ per cent of the indicated chart record to the value given by the chart when the cam is cut for a range of from 175 deg. to 225 deg., average 200 deg.

Correction for scales = 1.001875
Correction for temperature = 1.035

581,904 × 1.035	= 602270.64	chart weight as corrected
595,750 × 1.001875	= 596867.03	corrected weight by tanks
Difference	= 5403.61	recorder lead
Result	= 0.9 per cent	recorder lead

TEST ON V-NOTCH METER NEAR NEW YORK, APRIL 1912

28 It is to be regretted that the series of tests made in April 1912 on a 400,000 lb. per hour V-notch meter cannot be described in full in this paper; the results in brief form are as follows:

29 This series of tests differs from the two tests just described

TABLE 4 TEST MADE NEAR PHILADELPHIA, PA., AUGUST 9, 1912

Consecutive Time	Elapsed Time	No. of Tank	North Bottom Reading	North Top Reading	Net North	South Bottom Reading	South Top Reading	Net South	Temperature
10:25.10*									
10:30.10	:05.00	1				26200	410	25790	
:35.07	:04.97	2	26250	100	26150				72
:40.09	:05.02	3				25875	150	25725	
:45.54	:05.45	4	26150	210	25940				72
:50.54	:05.00	5				25950	170	25780	
:55.64	:05.10	6	26150	220	25930				72
11:00.74	:05.10	7				26160	230	25930	
:05.87	:05.13	8	26330	220	26110				72
:10.96	:05.09	9				26220	210	26010	
:16.06	:05.10	10	26025	220	25805				72
:21.17	:05.11	11				25950	220	25730	
:26.29*	:05.12	12	26025	200	25825				73
:31.41	:05.12	13				26100	180	25920	
:36.53	:05.12	14	26075	200	25874				73
:41.66	:05.13	15				26030	200	25830	
:46.87	:05.21	16	26300	200	26100				
:52.00	:05.13	17				26025	160	25865	73
:57.20	:05.20	18	26000	180	25910				
12:02.39	:05.19	19				26450	160	26270	73
:07.53	:05.14	20	26000	210	25790				
:12.68	:05.15	21				26030	190	25840	73
:17.88	:05.20	22	26025	180	25845				
:22.96*	:05.08	23				26000	220	25780	74
1:57.86	:117.86		287420	2140	285280	312990	2520	210470	74
					310470				
			Total weight.....		595750				

* Marked chart and started to fill south tank.

in that the water was actually weighed on large platform scales, and hence no correction for temperature was necessary.

5-Hour Test on V-Notch Meter, April 19

Water by actual weight, lb	291965
Water by recorder, lb	290525
Lead of scales, lb	1440
Lead, per cent	0.49

8-Hour Test on V-Notch Meter, April 20

Water by recorder, lb	267260
Water by actual weight, lb	266009
Lead by recorder, lb	1251
Lead, per cent	0.46

13-Hour Test on V-Notch Meter, April 22

Water by actual weight, lb	557974
Water by recorder, lb	557775
Lead of scales, lb	199
Lead, per cent	0.03

8-Hour Test on V-Notch Meter, April 23

Water by actual weight, lb	352725
Water by recorder, lb	351501
Lead of scales, lb	1224
Lead, per cent	0.34

8-Hour Test on V-Notch Meter, April 24

Water by actual weight, lb	352725
Water by recorder, lb	352231
Lead of scales, lb	494
Lead, per cent	0.14

CONCLUSION

30 In conclusion it will therefore be noted that the V-notch method of measurement may be consistently depended upon within 1 per cent, even in view of fluctuating conditions of the rate of flow and also fluctuating temperatures. By using $\frac{1}{4}$ 90-deg. notch weir, still greater accuracy may be obtained, and for all particularly accurate tests either $\frac{1}{2}$ 90-deg. or $\frac{1}{4}$ 90-deg. weir plates are recommended.

31 Attention is here called to an able discussion of the V-notch with results of tests made by James Barr, B.Sc. Carnegie Research scholar at the James Watt Engineering Laboratories, Glasgow University, 1907–1909. The synopsis of his results appear in London Engineering, April 8 and 15, 1910 editions.

DISCUSSION

W. S. Giele told of tests made under his supervision by the Harrison Boiler Works on a Cochrane independent meter which was equipped with a Lea recorder; these tests were made because at that time there were no data available in this country in connection with meters of large capacity. As to

the correction for temperature of the water during the progress of the test, the speaker stated that no such correction is made because the V-notch is a volume machine, and the transformation from volume to weight is made in the cam shown in Fig. 3. But there is a temperature correction necessary between the temperature for which the cam is cut, and the temperature at which the measuring tanks were calibrated (in this case 3.5 per cent). It is a peculiar fact that this kind of instrument is to a large extent self-compensating as the submergence of the float varies with the density of the water, and thus in a measure compensates for the variation in the volume of the water going through the notch. In some tests made by the speaker the error was as little as ½ of 1 per cent.

The V-notch weir referred to by Mr. Yarnall is not in itself new, its formula having been investigated and developed by Thompson in 1858 and also investigated with more refined apparatus by Barr in 1907. The more general adoption of the V-notch weir has been handicapped by the lack of a recording apparatus sufficiently delicate to take advantage of the superior accuracy of the V-notch weir, particularly at low rates of flow and Mr. Yarnall's paper describes a recording apparatus which is intended to meet this need.

J. H. Browne described how the hook gage was applied to obviate the large expense and time necessary to test the type of meter used in the Dartmouth College tests. In Table 2 can be seen a hand-hole on the front side of the V-notch. In conjunction with the engineer at the plant the speaker inserted a hook gage in this hand-hole sufficiently far back from the V-notch so that it would not be disturbed by the flow of water, and took observations on the hook gage every three minutes for one hour and thirty minutes. Simultaneously the engineer took observations on the vertical scale in the case of the recorder. These observations were found to coincide with each other to the third decimal, in inches, that is to 0.001 in. By this and similar methods it was also proved that the meter is correct over a very fluctuating flow.

Fred. N. Connet thought that a wrong inference would result from the statement in Par. 13, that the accuracy of a venturi meter or pitot tube depends upon *velocity changes* and that fluctuations in velocity therefore impair the accuracy. While it

is true that the differential pressure at the two points in a venturi tube is due to the difference in velocity at those two points, it is not true that it is due to any changing or varying in that velocity. Any inaccuracy due to a changing or fluctuating velocity is just as sure to apply to a V-notch as it is to a venturi or pitot tube. In the device described by Mr. Yarnall, the large tank exerts a steadying effect upon the velocity of the water and credit should not be given to the notch but to the tank. A tank used in series with a venturi tube gives exactly the same beneficial results, although such a tank is seldom required.

When installing a boiler feed meter of any type whatever it is customary to select one having a maximum capacity about twice the ordinary load. It is desirable therefore to have the greatest accuracy of the meter apply to rates which are approximately one-half the maximum capacity. The V-notch is more accurate for low velocities because there is a considerable movement of the float corresponding to fluctuations in the flow at that point. As the water, however, rises in the V-notch, the width of the weir becomes greater and the float movement corresponding to variations in velocity becomes less and less. As the paper states, a 90-deg. V-notch weir with a capacity of 100,000 lb. per hr. has a maximum float movement of 6 in., and at a 50,000 lb. rate the float movement is about 4½ in. At this rate a variation of 1/16 in. in the float elevation changes the meter readings about 2000 lb. per hr., equal to about 4 per cent of the nominal capacity. The error due to a departure of the float from its intended position will be relatively smaller in meters of greater capacity, and vice versa, but the capacity mentioned probably represents an average of the boiler feed meters in use in this country. In the venturi meter the total float movement is about 22 in. for meters of all capacities, and when running at half speed a variation of 1/16 in., which is more than would probably occur, corresponds to only 3/10 of one per cent of the nominal capacity.

THE AUTHOR said that he had been working for some time on a compensating float of a varying section such that, should the density of the water change owing to changes in the temperature, it would correct exactly by weight for those changes due to its being immersed more or less. When a V-notch meter is used for measuring hot water, as the temperature rises the buoyancy decreases, and the float which is the actuating means for

the recorder drops a little in the water. Over a range of about 50 deg. it compensates very closely indeed. The float used is a double conical float, with a 3-in. straight barrel. It was found that with the float so formed an almost complete compensation for the changes of temperature could be obtained. The tests in this direction are not finished yet.

The test referred to by Mr. Giele was made on a Lea V-notch recorder having a maximum capacity of 300,000 lb. per hr. It is interesting to note by referring to the data of this test in the paper that the compensation for changes in temperature was almost exact. The only correction necessary was for the temperature at which the test was run, namely 73 deg., while the cam of the Lea recorder was cut to measure water at 212 deg. This correction was predetermined from the known formula of the flow, used in plotting the cam for the drum.

No. 1375

RECENT DEVELOPMENT OF GAS POWER IN EUROPE

By H. J. Freyn, Milwaukee, Wis.

Chairman of the Gas Power Section

The development of the internal-combustion engine in Europe during the last year has been very marked and great strides were made, especially by the Diesel engine. The unparalleled progress of this prime mover, due primarily to the expiration of the basic Diesel patents, created a wave of enthusiasm, which swept over Europe in full-tide only a short time ago. To the careful observer, however, signs of an ebbing of this wave are apparent and more sober, more healthy views are gaining ground.

OIL ENGINES

Large Diesel Engines.—Without desiring to detract from the importance of the Diesel engine, investigation has proved to me that my warning against over-enthusiasm sounded about a year ago on a similar occasion was fully warranted.

In particular, I was disappointed to find that the achievements with large Diesel engines, above 1000 h.p., do not altogether coincide with the optimistic reports which have appeared in the technical press of the world during the past year or two. It is not certain that the large Diesel engine will be an undisputed commercial success abroad, since the price of natural fuel oils is steadily increasing, thus largely limiting such engines to the use of tar oil.

The horizontal double-acting four-cycle M.A.N. Diesel engine in twin tandem arrangement of 1600 to 2000 b.h.p., which I had the privilege of examining at Halle a. S., is a splendid piece of machinery and operates apparently very satisfactorily on tar oil with a small

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addition of ignition oil. It is claimed that such an engine can be sold in Germany for from $30 to $32.50 per horsepower, against $35 per horsepower for a steam plant. It should be expected, however, that the repair and maintenance cost would be much higher than for a steam plant and even higher than for present gas engines, on account of the high pressures and temperatures which are essential with the Diesel working principle and because of carbonization at partial loads which can be prevented only by scrupulous cleanliness. In countries where the price of coal is high, this engine type may have some importance, and a number of such engines of even larger unit capacity than mentioned have been sold in France and Russia, where fuel conditions are favorable for this type.

Small and Medium Sized Diesel Engines.—The Diesel engine in sizes of from 40 or 50 h.p. in single-cylinder units, up to about 600 h.p. in four-cylinder units, plays a vastly more important part abroad. It has unquestionably come to stay and has reached a high degree of perfection which places it in line with the corresponding steam or gas-engine plant as far as reliability and cost of operation are concerned, and far ahead of its competitors from the standpoint of fuel economy. It is amazing to note how many manufacturers of gas and steam engines abroad have taken up the manufacture of Diesel engines, because they found that the sale of suction gas producer plants and smaller steam engines had fallen off alarmingly during the last few years.

The reason is plain. The single-acting, four-cycle, single or multi-cylinder Diesel engine, but particularly the former, is comparatively simple in construction and operation, especially when its unequaled efficiency is considered, for it is certain that, generally speaking, complication of any mechanical contrivance increases with increasing efficiency. This engine is always ready to be put in service at a moment's notice and has no standby losses; it does not require upkeep and attendance of boilers or gas producers and its cost compared with that of a steam or gas plant is reasonable. It can be installed in the basements of buildings below occupied dwellings, whereas in Europe boilers are not permitted in such places. One of the greatest advantages which can not easily be expressed in figures, but which is of vast importance to the consumer, is the fact that the actual fuel consumption of Diesel engines, taken over long periods of operation, does not materially exceed the guaranteed figures, whereas in gas producer and steam plants this excess is quite considerable. A few months ago a German technical periodical contained an article

showing that the actual fuel consumption during a period of one whole year was for certain Diesel engine plants only 14 per cent higher than the guaranteed figures given by the manufacturer, whereas it averaged 120 per cent more in high-class steam locomobile plants. In this comparison 27 installations containing units of 50 to 80 h.p. were investigated, 12 of which were locomobile plants, whereas 15 contained Diesel engines. In a Diesel plant the human element, the skill of the operator, must have much less influence upon the fuel economy than in a steam or producer gas plant where everything depends upon the efficiency and intelligence of the fireman and producer attendant.

While the Diesel engine was originally a vertical engine, the last two or three years have witnessed a remarkable development of the horizontal Diesel engine. With few exceptions the manufacturers of vertical Diesel engines have branched out to build horizontal engines as well for purely commercial reasons. Experience shows that the fuel consumption is only very little, if any, higher than that of vertical engines, but the horizontal type is decidedly preferred by the customer, not only on account of its lower cost, but particularly owing to its greater simplicity and better accessibility. Especially the horizontal Diesel engine with Lietzenmayer's open fuel injection nozzle offers these advantages in their full measure. The open nozzle is simpler, can be more easily cared for by unskilled attendants, gives better results when burning heavy oils and is less sensitive in its action than the so-called closed injection nozzle of the original Diesel engine. The horizontal open nozzle oil engine has the additional advantage that certain essential accessories, such as the starting gear and the fuel pump, can be designed much simpler than for the other type. Many engines of the former type are in very successful service, using all kinds of fuels and installed for various purposes, for instance, for power transmission, for operating flour mills, for lighting factories, sanitariums, office buildings, department stores and for electricity works.

The only difficulty of which some users of small and medium Diesel engines sometimes still complain is caused by the discharge valves of the air compressors for fuel injection, and although these difficulties have been greatly reduced, numerous efforts are being made to materialize the dream of the "compressorless" Diesel engine. I had the privilege of seeing a 12-h.p. experimental engine of this type in operation, which seemed to give good results, since an oil consumption of only 0.48 to 0.5 lb. per b.h.p-hour is claimed, but it

is not certain that this principle, which has been tried many times before, would be applicable for larger engines.

Marine Engines of the Diesel Type.—In view of the great importance of the Diesel engine for ship propulsion, there is little wonder that great effort is directed by many manufacturers to the design and construction of an engine which, while having nearly the same excellent fuel economy as the stationary engine, will also meet the severe requirements of simplicity, reliability and reduced first cost. That the last condition is not yet fulfilled is shown by the fact that, according to authentic information, the cost of an 800-h.p. four-cylinder, two-cycle marine engine recently built was 50 per cent higher than anticipated and 150 per cent higher than for a corresponding steam engine equipment. I was nowhere able to find the 10,000- and 15,000-h.p. Diesel engine ships of which I had read, and even enthusiasts do not believe that our modern leviathans will be propelled by Diesel engines.

The largest single Diesel engine unit now propelling any ship develops not over 2000 h.p. and there still remain many obstacles to be overcome before the single-acting multi-cylinder, four or two cycle Diesel engine on board ship is supplanted by the double acting engine. Vertical engines of the latter type are still in the experimental stage and their future does not appear very promising on account of the serious mechanical complications of scavenging pumps and multiplicity of scavenging valves in each cylinder head.

The lack of simplicity in these designs has prompted experiments with the so-called "valveless" two-cycle type, which presumably will become the large marine Diesel engine of the future. One valveless two-cycle experimental engine of 260 h.p. is claimed to have shown an economy as good as that of a regular two-cycle engine with scavenging valves. The mean effective pressure obtained reached 125 lb. per sq. in., which is truly remarkable in view of the extremely simple scavenging method employed.

The Junkers Oil Engine.—Among the valveless, two-cycle engines, the Junkers oil engine has attracted a great deal of attention. It has reciprocal pistons, a three-throw crankshaft, and differs essentially from the regular type. An advantage is the excellence of the form of combustion space which, from a thermodynamic point of view, at least, gives it the distinction of being the best oil engine. Practical experience alone will tell whether its unusual design and certain mechanical disadvantages will be offset by this advantageous feature. One of the foremost authorities in Europe very graphically charac-

terized the Junkers engine as an oil engine built around a given perfect combustion chamber in contradistinction to all other Diesel engines where it is attempted to provide a satisfactory combustion space in an existing engine. Only a few smaller engines, mostly for marine propulsion and of the twin or three-cylinder vertical type have been built abroad. The tandem arrangement of cylinders which results in the same number of crank efforts realized in ordinary single-cylinder, double-acting, two-cycle engines has so far been applied only on two 800-h.p. three-cylinder vertical engines to be fitted in a vessel of one of the largest steamship companies. Aside from two experimental engines in Professor Junkers' famous laboratory, no horizontal engines of this type have been built and opinions abroad are much divided concerning the future of the stationary, horizontal, tandem Junkers engine.

Conditions Affecting the Development of the Diesel Engine.—If conditions in Europe are not altogether favorable for the development of the large Diesel engine, they are much less so in this country. The misapprehension frequently prevails that the Diesel engine must have a wonderful field of usefulness in countries where crude oil is produced in very large quantities, but such is not the case. The price of oil fuel in any locality must be considered in connection with the prevailing coal price to determine whether a Diesel engine installation is economically efficient or wasteful. Cheap oil fuel will favor the Diesel engine only where coal is very expensive, as on the Pacific Coast, whereas high oil and low coal prices will exclude it from any part of the country where such conditions exist. Considering that the total cost of producing power is made up of a number of items, as interest on the money invested and depreciation of the plant, cost of fuel, cost of attendance, maintenance and repairs, cost of supplies, etc., it is evident that in Europe where fuel prices are from two to three times higher than in this country,while labor is only one half to one third as much, the item of fuel cost constitutes a very high percentage of the total cost of power, particularly since the other items, such as interest and depreciation, attendance and repairs, are actually much lower and thus form a still smaller percentage of the total cost.

With us conditions are practically reversed: Our fuel cost is not only lower and our labor cost much higher than in Europe, but, moreover, the quality of our mechanical labor is not as excellent, and thus the fuel cost plays a much less important role in the total cost of power than the items into which labor largely enters; further,

the lower standard of manufacture, and especially of maintenance, favors the use of simple although less economical machinery.

To what importance the use of coal tar oil has grown on the continent can be judged from the fact that France has to rely on this Diesel engine fuel almost exclusively on account of the very high import duty on natural oils. In Germany a number of collieries have combined to form a company for the utilization of tar which operates a large factory at Meiderich, where, at present, over 300,000 tons of tar a year is distilled, producing over 80,000 tons of tar oil. This is contracted for in larger quantities at a price of approximately $12.50 per ton (5 cents per gallon). With the tremendous increase in demand, it is expected that before long this one factory alone will have a capacity for handling one million tons of tar per year, producing 300,000 tons of tar oil.

It is estimated that the total world production of coal tar oil is even now in the neighborhood of 1,000,000 tons per annum, and since it is more and more recognized that coal is too valuable a fuel to be waste in our present-day furnaces the number of by-product coke oven and by-product gas producer plants will steadily increase, especially in our country, so that the total production of tar oil is certain soon to double and treble, adding to our supply of natural liquid fuels an artificial product which will by legitimate competition check the advance in the price of the former.

The heat value of tar oil is over 17,000 B.t.u. per lb. and its consumption in the Diesel engine is, therefore, not very much higher than that of natural oil, varying now from 0.49 to 0.46 lb. per b.h.p-hour, according to the size of the engine.

The operation of medium size Diesel engines on tar oil, especially of those above 50 h.p. per cylinder and with open fuel injection nozzle is satisfactory. For smaller engines the addition of a so-called ignition oil is necessary, which somewhat complicates these engines.

GAS TURBINES

Renewed activity is noticeable abroad in the matter of designing an adequate gas turbine. The foremost authorities are convinced that the gas turbine will soon come to stay and, in view of the universal tendency towards high-speed rotative machinery, the statement would seem warranted that when the gas and oil turbine does come it will revolutionize power production in the large steel centers and wherever natural fuel oils abound in this country. A large experimental gas turbine in Germany is operating splendidly, me-

chanically speaking, although the problem of satisfactory thermal efficiency still remains to be solved.

INDUSTRIAL WASTE GASES

The use of the so-called "industrial waste gases" has progressed considerably. The four most prominent manufacturers in Germany alone are now building large gas engines for blast-furnace and steel works at the rate of 120 per year and a number of twin-tandem engines of 5000 b.h.p. operating very satisfactorily on rich coke oven gas can now be seen in German coking plants.

The splendid results obtained with gas blowing engines in Germany have prevented the turbo-blower from getting a firm foothold in that country, although its use in France, England and Russia is steadily increasing. Blast-furnace gas engines for blowing and electric power purposes and even for operating rolling mills are today the standard equipment for old and new plants, even where the coal price is fully as low as in our Pittsburgh district. Within the last two years the use of coke oven gas for producing electric power by gas engines has greatly increased, and after overcoming the many earlier difficulties the coke oven gas engine is today just as safe and reliable as the blast-furnace gas engine. These difficulties arose principally from overrating the engines, and from sulphur in the engine gas. Since the reasons for the trouble were recognized, the causes, or their consequences, are now avoided, so that in a number of plants sulphur of considerably over one grain per cubic foot of gas is very successfully handled in coke oven gas engines if the proper very simple precautions are taken.

DETAILS OF CONSTRUCTION

In regard to the detail construction of the modern German gas engine a few remarks may be of interest. The most prominent gas engine builders are now governing their engines on a simple combination quantity-quality regulation principle. The valve gear has been much simplified by all manufacturers, and gas engine prices are, generally speaking, lower than they ever were before. The gas cylinders, after a few unsatisfactory attempts to use cast steel, are invariably made of cast iron and with the exception of a few types are cast in one piece with integral water jacket. They are often provided with hard cast-iron liners. The cracking of gas cylinders is now much reduced by proper design, selected material and better pouring methods. Experience has taught that the life of a gas cylinder can be materially increased if the water jackets are thor-

oughly cleaned from all scale and muddy deposit at regular intervals of from four to eight weeks, depending upon the character of the cooling water used. It was found by experiment that the average temperature of the cylinder wall increases at an amazing rate with even a very thin layer of scale forming in the water jacket, and in certain plants with hard cooling water, acid and wire brushes are resorted to from time to time to take the accumulated scale off the cylinder walls.

Gas cylinders on electric engines crack more frequently than on blowing engines because the former carry much higher loads than the latter and because electric engines are usually shut down oftener. It was found that frequent shutting down and starting of gas engines shortened the life of the cylinders materially if the engines were allowed to cool off too much during shut downs.

The average life of a gas cylinder in the large steel plants in Germany is pretty generally reckoned to be from five to six years. This is accepted as a matter of course and the expense incurred by replacement is simply distributed over this number of years in the cost figures.

The cost of producing electric power in German blast furnace and steel plants is about $2.70 per thousand kilowatt hours, comprising $1.25 value of the raw blast-furnace gas and including cost of gas cleaning, cost of labor, repairs, and supplies of all kinds, but without interest and depreciation on the money invested. This figure applies on an annual output of about 100,000,000 kw-hr. and a use factor of approximately 60 per cent.

The gas pistons are generally ribless iron castings, although for large engines cast steel is used. One concern has recently introduced for blast-furnace gas engines solid, non-cooled, forged steel pistons of one piece.

The piston rods are now usually made of open hearth steel of about 100,000 lb. tensile strength, uncambered, and their diameter is approximately 26 per cent of the diameter of the gas cylinders. A variety of piston rod coupling designs is used but the threaded connection enjoys the best reputation. It is noteworthy that crossheads are made of forged nickel steel.

No uniformity of design can be claimed for piston rod packings, but they invariably consist of single or double sectional cast-iron packing rings pressed against the rod by garter springs and separated by solid one-piece rings forming from 12 to 14 chambers. It is universally claimed that the clearances of the ring sections must be

reduced to the possible minimum. So-called fire rings are sometimes used, but several builders object to such solid rings.

The exhaust valves are made of forged steel or of Durana metal and are now almost exclusively of the non-cooled mushroom type. Some builders, however, still prefer water-cooled exhaust valves for coke oven gas engines.

GAS CLEANING METHODS

The progress of the large gas engine in Europe has been accompanied by great improvements in the method of purifying blast furnace gas. Until recently two systems of cleaning gas for engine purposes were largely in use, both of the so-called "dynamic" type, and by using considerable quantities of water: the hydraulic fan and the Theisen washer. Results obtained with these appliances regarding both cleansing efficiency and cost of operation are satisfactory. Nevertheless, they now are superseded in Europe by new processes, which still further reduce not only the cost of installation, but principally the cost of purification on account of the smaller power and water consumption.

One of these new methods is a radical departure from the former wet-cleaning processes and works on the principle of filtering the gas in a perfectly dry condition, so that no water is required save that necessary to cool the engine gas to atmospheric temperature. The filtering is achieved by means of Beth filter bags, which are made of special fabric and suspended vertically in several air tight compartments. The raw blast-furnace gas coming from the furnaces at high temperature is first cooled by radiation to about dewpoint (around 130 deg. fahr.) and thereupon superheated approximately 20 to 30 deg. fahr. by using any available source of heat, such as waste heat from gas engines, exhaust steam or even the sensible heat of the raw blast-furnace gas itself. By cooling the gas a large amount of moisture is removed and the scorching or burning of the filter bags prevented, whereas the subsequent superheating avoids condensation of the remaining vapor and clogging of the bags. The raw gas under the action of a fan passes from the inside to the outside of the filter bags, depositing all impurities on the surface. The layer of dust naturally increases continually and would eventually render the bags impenetrable, were it not for an automatic mechanism which in a fashion, similar to that used in flour mills, shakes the bags at regular intervals of about four minutes

and causes the dust to drop off. Simultaneously the compartment which is to be cleaned is automatically separated from the others, and the direction of gas current reversed; the purified gas surrounding the filter bags now enters the latter, opening the pores of the filter material, which were previously closed up by the passing dirty gas current. The dust shaken off the filter bags falls into a receptacle, from which it is removed in a perfectly dry condition.

Two experimental filter plants installed at Halberger Huette and Duedelingen have capacities of about 10,000 and 14,000 cu. ft. of gas per minute and clean the gas for gas engines only. It is claimed that the power consumption is 2.5 to 3 times lower than that of hydraulic fans and Theisen washers, that the water consumption is only one-fifth to one-sixth of the corresponding requirements of wet-cleaning plants, and that the dust content in the purified gas is below 0.0004 grains per cu. ft., or fifteen to thirty times lower than in the purified gas obtained with the older wet cleaning apparatus. The cost of operation is furthermore claimed to be lower and the cost of installation only a little, if any, higher.

As the advantage of using thoroughly cleaned blast-furnace gas, not only for gas engines, but for boilers and stoves is more and more appreciated, this "Halberger" System causes much interest and comment and several prominent blast-furnace men abroad assert that the cleanest blast-furnace gas obtainable at comparatively low cost is just clean enough even for boilers and stoves, which, after all, form precisely as important and costly a part of their installation as the gas engines. In certain districts of Germany and Luxemburg, water is exceedingly scarce, and the laws against river contamination are so strict that large sums of money have to be spent to purify the washing water before it can be disposed of. Setting tanks of huge area have to be installed, and the disposal of the mud presents difficulties and is very costly; furthermore, flue dust in a moistened condition is of no use except as filling in material and since it contains considerable quantities of iron ore, its disposal in such a way is rather wasteful.

On the strength of the satisfactory results obtained, many blast-furnace plants in the Minette District, and even in Westphalia, have decided to install gas filtering plants, varying in capacity from 17,500 to 140,000 cu. ft. of gas per minute. One

manufacturing concern alone, which acquired the rights for Germany, had in the month of July orders for twenty-one plants, with a total capacity of 825,000 cu. ft. of gas per minute and valued at one million dollars.

The other new system of gas cleaning makes use of the " disintegrator " principle. Two well known concerns in Germany, specializing in gas washing apparatus, have developed this system on somewhat different lines, and one of them claims to have on hand orders for disintegrator gas washers capable of purifying 1,750,000 cu. ft. of gas per minute. One machine only is used for preliminary purification of boiler and stove gas, from 2 to 3 grains of dust per cubic foot to 0.088 to 0.175 grains, with a water consumption of only 10 gallons and a power consumption not exceeding 0.072 h.p. per 1000 cu. ft. For gas engine purposes two disintegrators are placed in tandem which causes the power consumption to increase to 0.13 h.p., the water consumption to 18 gallons per 1000 cu. ft., whereas the amount of dust in the gas drops to 0.0044 — 0.0088 grains per cu. ft.

No change has taken place in the method of purifying coke-oven gas; bog or swamp ores are still used for sulphur cleaning, although one plant at least has succeeded in overcoming the disadvantage of occasional regeneration of this material by alternate exposure to the air by regenerating it continually.

GAS PRODUCER DEVELOPMENT

It seems extraordinary that in spite of the high price of coal the development of the gas producer for power purposes has not kept pace in European countries with that of the gas engine. For many years the Mond gas producer with sulphate of ammonia recovery has been well introduced in England, and recently much interest has been shown in Germany in such plants, which offer material financial advantages over ordinary gas producer installations. Since a great deal of sulphate of ammonia now finding a ready market as an artificial fertilizer, can be recovered from the coal, the cost of gas making is considerably reduced, and installations are known where the saving thus obtained reaches the figure of $2.00 per ton of coal, which in many cases is as much as, or more than the value of the coal itself. The gas manufactured in such plants has a heat value of about 150 B.t.u. per cu. ft., and is of very uniform quality; on account of the removal of tar and dust, it is particularly adapted for gas engines, but its low cost makes its use very profitable for all

kinds of metallurgical purposes, such as heating and annealing, for calcining, roasting, distilling, for glass works operation and a variety of other industries. Ordinary coals as well as low-grade fuels, such as peat, lignite, brown coal, etc., can be very successfully gasified in such plants.

COKE OVEN GAS

The growing demand for coke, which goes hand in hand with the soaring pig iron production of the world, has caused a remarkable increase in coke oven plant activity. The days of the wasteful "Bee-Hive" coke oven are fortunately past, and the highly economical by-product coke oven is gradually taking its place. The manufacture of coke entails the accumulation of large quantities of coke braise, which cannot always easily be disposed of. Successful attempts were recently made in Europe to gasify this coke dust in special gas producers. One system operates with liquid slag very much on the order of a blast furnace, whereas the other adapts a well known European producer design to the special requirements of gasification of very fine fuels. In a number of European blast-furnace and coke-oven plants, this coke braise will be gasified in such producers, and the resulting gas of about 150 B.t.u. per cu. ft. admixed to blast furnace gas, to increase its quantity and improve its quality.

Mixtures of industrial gases are used more and more for metallurgical and power purposes abroad; in one plant in England a mixture of blast-furnace and coke-oven gas is used for all heating purposes and to operate gas engines. At first glance it may seem difficult to obtain a sufficiently uniform mixture of two gases so different in specific gravity, but experience has shown that modern blast-furnace gas washers of the revolving type, such as fans, Theisen washers and disintegrators are splendidly adapted for this purpose, and that changes in the quality and heat value of the mixed gas of even considerable magnitude, do not seriously affect the operation of gas engines.

In this particular steel plant another very interesting novelty of perhaps great importance for future gas engine application is in operation, namely, a system of overloading four-cycle gas engines 25 to 35 per cent of their normal capacity.

NEW GAS ENGINE METHOD

The method devised and patented by one of the foremost gas engine manufacturers in Germany and which creates an almost

perfect analogy with the condensing plant of steam engines consists in scavenging ordinary four-cycle gas engines and introducing the fresh mixture under a pressure somewhat higher than atmospheric. This is achieved by raising the pressure of the combustion air and gas in turbo-blowers to a gage pressure of a few inches of mercury and by scavenging the spent gases of the previous power stroke with a blast of this air, subsequently introducing gas and air under slight pressure into the combustion chamber. By utilizing the waste heat of the gas engines for raising steam in suitable waste heat boilers and driving steam turbines direct-connected to the turbo-blowers, the power requirements of the auxiliaries can be amply supplied without resorting to outside sources of power, nor detracting from the increased output of the gas engine generator.

An overload of 35 per cent was attained without raising the initial pressure—an important point—and without appreciable increase of the heat absorbed by the cooling water; the thermal efficiency of the scavenged engine was found to be somewhat better and the mechanical efficiency considerably better than that of the non-scavenged engine.

In another steel plant two tandem engines of 1360 kw. or 2000 b.h.p. normal rating (at 65 lb. m.e.p.), which were provided with generators of 1700-kw. capacity in anticipation of the overload scheme, can carry loads of 1800 to 2000 kw.; before the changes were made in April 1912, the average kilowatt output per engine in twelve hours as ascertained from the log book was 13,000 kw-hr., but since installing the scavenging plant, the average output was raised to 18,000 and 20,000 kw-hr. in a twelve-hour run, corresponding to an average increase of approximately 40 per cent.

Regarding the financial aspect of the matter, it can be shown that the capacity of an existing gas engine plant can be increased 30 per cent, provided the generators are large enough, at a considerable saving compared with the cost of providing for this increase in capacity by installing additional non-scavenged units.

HUMPHREY GAS PUMP

The technical press has been rather silent during the last year regarding the Humphrey pump, but according to reliable information, this is by no means an indication that this gas pump has passed out of existence; on the contrary, one of the largest

concerns in Germany is devoting much time and effort to experiments on generating electric power with such a pump at its factory, and a 40,000,000-gal. pump will be installed in England with a guaranteed coal consumption of 1 lb. per effective pump horsepower.

DISCUSSION

William T. Magruder. I had the pleasure of spending a few months on the other side of the water this past summer, and looked at the subject of gas power from a different point of view from Mr. Freyn. I considered it both from the engineering and educational sides and received much information not only from manufacturers as to costs and details, but also from professors of engineering. In the first place, I noted, in both Germany and England, that the most important developments in gas and oil engines in recent years have been made by consulting engineers, who are graduates of universities and engineering colleges, many of whom are professors or lecturers who are doing much advanced work along the line of investigation. Secondly, that societies like the Institution of Civil Engineers and the Institution of Mechanical Engineers, the British Association for the Advancement of Science, and the Society of German Engineers, are deeply interested in the subject of gas power, and their members are doing the largest part of the scientific work. It is interesting to note the number of papers presented to these learned societies from men who have received higher degrees, such as Master of Arts, Doctor of Science, and Doctor of Engineering. I would respectfully recommend their activities to the attention of The American Society of Mechanical Engineers, and the American Association for the Advancement of Science, as it is high time that they too become more active and that our Research Committee on Gas Power did something in the way of scientific research in the field of gas and oil engines.

Few Americans appreciate the tremendous difference between the gas and oil engine industry in Europe and in America. The price of gasolene in Great Britain is 40 to 45 cents for an Imperial gallon, which is the equivalent of 33 to 37 cents for the American gallon, or double the price, as compared with 16 or 18 cents paid for the same amount in the United States. It can be readily seen why Englishmen are now looking for a cheaper fuel than gasolene, as Americans will be shortly doing, and why oil engines operated by tar oil are being used especially

in Germany, although it is a much less desirable fuel than other oils on account of the difficulty in igniting it. It can be seen why gas producers operated by the combustion of peat briquets in Scotland and Germany and bituminous gas producers are favorably regarded. Because Dr. Mond happened to live in England, the Mond producer is more commonly used in England than in Germany. It used to be that 1500 h.p. was the smallest unit that could be economically installed and used. I was told by one manufacturer in England that now they could be economically used in England in plants as small as 800 h.p. One of the interesting and surprising things was to see the large numbers of gas and oil engines on the testing floor and of producers in the assembly buildings, to go through the painting and shipping rooms and to be told that the output of several of the works visited was from 2000 to 3000 gas and oil engines per year and 30 producers a month. The reason for this, it was stated, was the low price of city gas and the relatively high price of coal.

The smaller proportionate number of motor cars used in England as compared with this country is doubtless due to the relative prices of suitable fuel. I was informed that there are 150,000 motor cars in Great Britain for a population of fifty million people, or about three cars per 1000 of inhabitants. In this country, almost 1,000,000 motor cars were registered in 1912, to say nothing of the motor cycles, motor boats and other users of oil power; or one car for each 100 of the inhabitants of the United States. But to this large number of motor-car engines must be added more than 1,000,000 gasolene engines used for farm work and in motor boats in this country. Taking the average horsepower of the motor cars to be 25 and of the motor boats and farm engines to be 10, thirty-five million horsepower of gasolene engines are immediately available for the practical generation of power in this country. Suppose they used as a minimum at rated load ¾ lb. or 1 pt. of gasolene per h.p. per hr. for 1 hour of use they would generate thirty-five million horsepower and require four and a half million gallons of gasolene at an estimated cost of $700,000. The total annual supply of American gasolene is estimated to be one billion, five hundred million gallons, and that would last only 333 hours. If every gasolene engine were run at its rated load each day, the annual distillation of gasolene in this country would be sufficient to permit them to be operated for only one hour per day, or 333

hours per year. In other words, our present annual output of gasolene is sufficient to operate continually at their rated load only 5 per cent of the gasolene engines now sold and in operation.

These being the facts as I gather them from statistics published by the Bureau of Mines, and other reliable sources, it can readily be seen to be a case of supply and demand. It is probable that some other fuel than gasolene must come into use for oil engines and that this fuel will be kerosene. When the vast amount of kerosene now on the market is realized, three billions of gallons being distilled annually, it seems certain that kerosene oil will come into very great and general use in the next few years.

It was interesting to note how the kerosene oil engine is being introduced and used in England and on the Continent in place of the gasolene engine and particularly in the form of the so-called semi-Diesel oil engine, and from which results below ½ lb. of kerosene per h.p. per hr. have been reported by expert engineers. This is a branch of the gas power field that Americans are only beginning to cultivate with any degree of efficiency.

The figures that I obtained abroad in respect to the fuel consumption of Diesel engines may be of interest. The actual fuel consumption of a large plant on test was 0.44 lb. of oil per b.h.p-hr. at full load, 0.45 lb. at three-quarter load, 0.47 lb. at one-half load, and 0.62 lb. at one-quarter load. Other figures obtained in tenders for alternating-current electrical units of 200 to 300 kw. capacity running on crude fuel oil having a calorific value of 18,000 b.t.u. per lb., varied for four-cycle Diesel engines from 0.603 to 0.662 lb. at full load per kw-hr., from 0.64 to 0.70 lb. at three-fourths load, from 0.70 to 0.83 lb. at one-half load, and from 1.14 to 1.27 lb. of oil per kw-hr. at quarter load. These figures confirm Mr. Haynie's remarks upon this point.

With respect to the first cost of Diesel engines, Mr. Freyn has given $32 to $35 per h.p. The lowest price given me in England on Diesel engines was $35 per b.h.p. and from $35 to $45 seemed the usual price in the larger units.

In the tenders above referred to for Diesel units of 200 and 300 kw. capacities, the prices for the engines directly coupled to alternating-current generators varied from £13-16-4 for the 300-kw. units at 192.5 r.p.m., to £22-17-3 for the 200-kw. units at 165 r.p.m., and the average price of 23 four-cycle Diesel engines and alternating-current generators of 200 and 300 kw. capacities from seven builders averaged £17-7-11 per kw. of rated capacity.

These are the equivalents of $67, $111, and $85 respectively. The horizontal engine units were about $15 per kw. less in price than the vertical ones. The corresponding figures for the gas-producer plant with engines and generator without foundations were £19, or $92, installed.

As to the Diesel locomotives, the one I saw would be simply prohibitive on account of first cost, labor in operation, and a number of other difficulties, which may yet be overcome.

The whole subject of oil power at the present time reduces itself to costs of fuels and to the usual commercial and engineering problems. If our commercial friends would be more fearless and less secretive and our scientific friends would be more energetic, the cause of the Diesel and other forms of oil engines, and of gas power in general, would be advanced.

W. R. Haynie (written)[1]. It may be generally stated that all heavier oils, if free from non-combustible foreign matter, can be readily used in Diesel engines. The non-combustible foreign matter is sand or mineral matter that will not burn, but will cut the engine cylinders; it can always be removed either by precipitation or by forced straining, and there is no petroleum residue of however heavy specific gravity, the ignition of which is not attainable by the temperature created by 30 to 35 atmospheres of compression. The complete combustion of the fuel oil in the cylinders after ignition is, however, entirely dependent upon the process of handling the fuel before it enters the cylinders. The period during which the oil has to be consumed is very short, only a small fraction of a second, and therefore the process of combustion must take place almost instantaneously. The rapidity with which the oil is consumed depends not upon its specific gravity or chemical properties, but upon the size of the particles or globules into which it is broken before coming in contact with the ignition temperature. The combustion proceeds over the surface of the particles, and the smaller the particles, the more quickly they are consumed. But when heavy residues are used in the Diesel engine without reducing the size of the globules, only the surfaces of the particles are consumed in the short time allowed for combustion, with the result that the interior portion is converted into a pitchy substance which deposits on the surface of cylinders, pistons and valves,

[1]U. S. Representative, Carels Frères (Ghent, Belgium), 30 Church St., New York.

producing waste of oil and clogging of the cylinder and valves. The oil entering the cylinder should be completely atomized.

The resultant effect of an oil used in the Diesel engine depends entirely upon its heat value, and therefore with the same heat units per lb. an oil of 0.97 specific gravity will produce the same energy results as a lighter oil of 0.85 specific gravity.

The speaker further stated that although he has read now and then in the papers of 6000 and 12,000-h.p. Diesel engines built abroad, he found when he went to Europe that they were still all on paper; engines up to 3000 h.p. may, however, be built quite successfully. The speaker used several lantern slides to illustrate his remarks.

In connection with Mr. Haynie's discussion it was stated by one of the members that Baron Schmidt had taken out patents on a two-cycle valveless engine which is now built by Cockerill Company at Seraing, Belgium.

Professor Magruder, in replying to a question of Mr. Chubb's as to the future possibilities of alcohol engines, said that alcohol engines in Europe are not a commercial success. The theory has been fully worked out, but as long as alcohol is heavily taxed and the trade throttled by commercial agreements, no real development can be expected.

D. G. Baker remarked that the only way of producing alcohol at the present time is from some vegetable growth, and all that can be produced from the soil is required to feed the men and animals upon it. It is possible to use some waste products, e.g. waste molasses, to manufacture alcohol, but in comparison with the large volume that would be required for the supply of internal combustion engines, any such production is only a drop in the bucket. Mr. Baker also raised a question as to the exact amount of tar that can be produced from a ton of coal. This, according to the speaker, is an important matter, because coal tar is a smokeless fuel that might help to conserve the fast diminishing supply of anthracite. In that direction a source of relief for our future oil production must be looked for.

E. Y. Sayer supplied the information desired by stating that in this country 10 gal. of tar oil are obtained per ton of coal of 2000 lb. A member stated in this connection that where the production of tar is the main purpose of the operation, an even larger production has been secured by means of special processes, such as distillation at low temperatures, etc.

No. 1376

MEASUREMENT OF NATURAL GAS

By Thos. R. Weymouth, Oil City, Pa.

Member of the Society

One of the most important problems encountered in natural gas engineering is the accurate measurement of gas in large volumes. For the purpose of discussion, the subject can be divided into two general classifications, namely, open flow and closed flow measurements. The first relates principally to the determination of the volume of gas a well is capable of producing, while the second refers to all closed pipe flows.

2 Under the first head, there are two general methods in use for measuring the open flow of a well, that is, the quantity of gas the well is capable of producing against an absolute back pressure of one atmosphere. The first of these is purely approximate, and is called the "minute pressure" measurement, wherein the well is permitted to blow into the air until its "head" is blown off, and is then suddenly closed in by a quick-acting gate, the accumulated pressure being noted at the end of one minute from the time the gate is closed. Then, Q, the open flow capacity of the well in cubic feet per day, on an atmospheric pressure base of 14.4 lb. per sq. in. absolute, becomes

$$Q = 1440\ V\left\{\frac{p_m + P_o}{P_o} - \frac{P_o}{P_o}\right\} = 1440\ V\frac{p_m}{P_o} = 100\ V p_m \ldots\ldots\ldots [1]$$

in which

V = volume of well tubing in cu. ft.

p_m = minute pressure in lb. per sq. in. gage

P_o = atmospheric pressure in lb. per sq. in. absolute

3 This equation assumes that while the gas is flowing freely into the air, the well is filled with it at a uniform pressure of one atmosphere, and at the end of one minute it is filled with gas at a

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uniform pressure equal to the minute pressure. Neither of these assumptions is correct, since in both cases there is a gradually increasing pressure from the top of the well towards the bottom, similar to that in a pipe line.[1] The resulting errors tend to counteract each other, however, and if care is taken in making the measurements a comparative idea will be obtained of the ability of the well to produce gas against a back pressure equal to a mean between the atmosphere and the minute pressure. Since the open flow will be greater than the flow against this back pressure, the assumption that the minute pressure gives an open flow measurement errs on the side of conservatism.

4 The second method of open flow measurement is that by means of the pitot tube, and is the one generally used for accurate determinations of well capacities. This method has been described in detail by Prof. S. W. Robinson,[2] who was probably the first to use it in connection with gas. The method was further discussed by him in the Geological Reports of Ohio,[3] wherein tables are published by means of which the flow may be computed from the pitot tube observations. These tables have been in general use by natural gas men for many years, and are based on the formula

$$Q = 1{,}462{,}250\ d^2 \left\{ \left(\frac{P_1}{P_o} \right)^{0.29} - 1 \right\}^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots [2]$$

5 Equation [2] was derived from the formula for adiabatic flow

$$v^2 = \frac{2gn \times 144\, P_o}{(n-1)\,\delta} \left\{ \left(\frac{P_1}{P_o} \right)^{\frac{n-1}{n}} - 1 \right\} \quad \ldots\ldots\ldots\ldots [3]$$

In equations [2] and [3]

v = velocity of flowing gas, ft. per second
P_o = absolute pressure of atmosphere, lb. per sq. in.
n = ratio of specific heats
δ = weight per cu. ft. of gas at pressure P_o
G = specific gravity of gas, air = 1.0
P_1 = absolute pressure shown by pitot tube, lb. per sq. in.
d = internal diameter of well mouth, in.
Q = open flow capacity of well, cu. ft. per 24 hr.

6 In computing his tables, Professor Robinson used the following values for these quantities:

[1] Problems in Natural Gas Engineering, The Journal, Am. Soc. M. E., May 1912, p. 723, fig. 5.

[2] Van Nostrand's Engineering Magazine, August 1886.

[3] Vol. 6, 1888.

$$n = 1.408$$
$$2g = 64.3$$
$$P_o = 14.6$$
$$G = 0.6$$
$$\delta = 0.0807 \frac{T_o}{T} G$$
$$T = T_o = T_s = 492 \text{ deg.}$$

where

T_o = absolute temperature of melting ice
T = absolute temperature of flowing gas
T_s = absolute temperature of storage

all being in fahrenheit degrees.

7 The ratio of specific heats for natural gas is more nearly equal to 1.266 than to the value used above. Inserting this value and using values for the other quantities as follows:

$T = 500$ deg.
$T_s = 520$ deg.
$P_o = 14.4$
Storage Pressure, $P_s = 14.65$

the formula becomes

$$Q = 1{,}758{,}560\ d^2 \left\{ \left(\frac{P_1}{14.4} \right)^{0.21} - 1 \right\}^{\frac{1}{2}} \ldots\ldots\ldots\ldots\ldots [4[$$

8 At 60 lb. gage pressure per sq. in. (the highest given by Professor Robinson), formula [4] makes the flow about 1 per cent lower than that derived from formula [2], whereas at 1 lb. it gives a result about 4 per cent higher.

9 The open flow measurement is not required with the same degree of accuracy as that of closed flow, where gas is being bought and sold, and at best is only intended to give a fair idea of the production. Even if an accurate measurement were made it would give no precise notion of the real productive value of the well when it must feed into the line against a high back pressure. Consequently, if it is desired to know the capacity under these conditions it is best to insert a pitot tube into the lead line from the well while operating under normal conditions. This involves the second division of the subject, namely, closed flow methods of measurement.

10 In closed flow measurements of gas where the volumes are small and the pressures are low, the ordinary positive gas meters are satisfactory, but for wholesale measurement some other type of meter must be adopted because of the prohibitive size of the positive type necessitated by large flows. The matter is further complicated

by the fact that usually when gas is bought or sold in large quantities it must be measured under varying pressures, necessitating the use of a correction factor to reduce the meter readings to quantities on a standard pressure base. There are many different types of proportional meters in use for this purpose, equipped with recording gages. The meter registration is then multiplied by a factor dependent upon the record of the gage in order to correct it to volumes based on the proper buying or selling pressure.

11 In addition to proportional meters there are a number of other devices in use which aim to attain a higher degree of accuracy, among which the most prominent are the venturi meter, the Thomas electric meter, the orifice, and the pitot tube.

12 The venturi meter is used principally for low pressure measurements, and while extremely accurate, is handicapped by possessing a complicated formula for the expression of volumes, which renders very tedious the computation of results from its records, and which demands a knowledge of the ratio of specific heats of the gas at constant pressure and constant volume, respectively. This formula is as follows: [1]

$$Q = 210{,}840\, A_2 \frac{T_s}{P_s} \sqrt{\frac{n}{(n-1)\,G}} \frac{P_1}{\sqrt{T_1}} \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \sqrt{\frac{1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}}} \quad \ldots\ldots [5]$$

wherein

Q = flow, cu. ft. per hr., on absolute pressure and temperature bases P_s and T_s
n = ratio of specific heats
G = specific gravity of gas
A_1 = area of venturi meter at entrance, in sq. ft.
A_2 = area of venturi meter at throat, in sq. ft.
P_1 = absolute pressure at entrance, lb. per sq. in.
P_2 = absolute pressure at throat, lb. per sq. in.
T_1 = absolute temperature at entrance, deg. fahr.
T_2 = absolute temperature at throat, deg. fahr.

13 For any particular meter and quality of gas, and assuming definite values of P_s, T_s, and T_1, equation [5] reduces to the simpler form

$$Q = KP_1R^a \sqrt{\frac{1-R^\beta}{1-CR^{2a}}} \quad \ldots\ldots\ldots\ldots\ldots [6]$$

[1] See Appendix No. 1.

where

$$K = 210{,}840\ A_2 \frac{T_s}{P_s} \sqrt{\frac{n}{n-1} \frac{1}{GT_1}}$$

$$C = \left(\frac{A_2}{A_1}\right)^2$$

$$R = \frac{P_2}{P_1}$$

$$a = \frac{1}{n}$$

$$\beta = \frac{n-1}{n}$$

14 The Thomas electric meter [1] is an extremely accurate instrument for the measurement of gas and is comparatively simple in construction. It operates on the principle that the energy required to heat any quantity of a given gas through a definite temperature range is directly proportional to the weight of the gas heated, and can therefore be used as a measure of it. In practice, electrical energy is used to supply the heat and is automatically controlled in such manner as to cause the temperature increase of the flowing gas to remain fixed at a value of about 2 deg. fahr. The electrical energy required to do this is measured by means of wattmeters, the dials of which can be graduated to read direct in cubic feet of gas based on any desired standards of pressure and temperature. The method is independent of the actual temperature and pressure of the gas entering the meter, depending solely upon its specific heat at constant pressure. Since the records of the device are obtained through the medium of wattmeters, they can therefore be made to give either a rate reading of flow, an integrated or totalized reading, or both.

15 This meter has been subjected to very thorough tests in conjunction with the pitot tube and venturi meter [1] which have firmly established the accuracy and reliability of all three of the devices.

16 The Thomas meter requires a supply of electric energy equivalent to about 1 kw-hr. for each 75,000 cu. ft. of free gas. This fact greatly restricts its use in the measurement of natural gas, especially in the field where such power is not usually available.

17 The most practical devices known to the author for general use in the accurate measurement of gas are the pitot tube and the

[1] Measurement of Gases, Carl C. Thomas, The Journal of the Franklin Institute, November 1911.

orifice, both of which are extremely simple in construction and are based on fundamentally correct principles. The formula expressing the flow of gas is identical for the two instruments for the range through which they are usually used.

18 The pitot tube was first described before the French Academy of Sciences in 1732 by its inventor, Pitot, who stated that it followed the principle of the law of falling bodies, namely $v^2 = 2\,gh$. For over a century after its discovery this wonderfully simple and accurate device was regarded merely as a scientific toy, until about 1850, when D'Arcy adapted it to closed pipe flows by the addition to the bent tube of a static pressure tube. It was then found that the difference in height of liquid in the two columns of a U-gage connected to the dynamic and static tips of the tube respectively, gave a true measure of the velocity, and followed Pitot's originally stated principle exactly, viz., $v^2 = 2\,gh$.

19 Prof. S. W. Robinson seems to have been the first to have applied the pitot tube to the measurement of gases. In 1873 he experimented with it in determining the velocity and form of jets of air from orifices, which led to the development of a commercial pitot tube for gas measurement.

20 If a pitot tube is inserted in a pipe conveying gas and a traverse made over the cross-section of the pipe, it will be found that the highest velocity exists at the center and the lowest at the walls. For this reason it has been customary for experimenters to determine the particular point in the cross-section of the pipe where the velocity corresponds to the mean velocity of the whole cross-section, and to keep the tube at that point for the determination of the quantity of flow as computed from the velocity readings. This point varies according to the condition of roughness of the interior surface of the pipe in which the tube is inserted, and accordingly it is necessary to determine it for each particular pipe used.

21 When a tube is to be used as a permanent gas meter, it is better, because of the fact just noted, to construct the pipe with extreme care, having the interior of uniform size and highly polished, and then to place the tip in the exact center of it. This necessitates the inclusion in the theoretical formula of a factor which in this paper is termed the "efficiency" of the tube, in order to reduce the flow, as indicated by the tube readings, to the value it would assume were the tube at the point corresponding to mean velocity instead of being at that of maximum velocity.

22 The tubes used by the author are all constructed in this man-

ner. The instrument as a whole is termed a pitot tube (comprising both the pipe and the bent tube inserted in it), and the bent tube portion of it is designated as the "tip." The pipe consists of 12 ft. of seamless drawn brass tubing with highly polished interior surface and gaged to accurate size. About 27 in. from the outflow end a brass saddle is sweated on, through which the tip projects into the tube, and which also contains the static pressure connection, which is merely a hole drilled through the saddle and tube flush with the inside wall. The tip is constructed in such manner as to be readily removable for cleaning, and yet exactly central in the tube when re-inserted.

23 A series of standard tubes of this type are maintained, against which all new tubes are calibrated. These standards, ranging in size from 2 in. to 6 in., were themselves calibrated by exhaustive tests against a large gas holder under different conditions of flow and pressure. The coefficients thus obtained were found to be constant throughout the complete range of water readings under which the tubes are used and also for all pressures.

24 The accuracy of these standards is attested by the paper by Professor Thomas already referred to, in which are given the results of a 45-day test at the Brave pump station of the People's Natural Gas Company of Pittsburgh, during which the Thomas electric meter was connected in series with pitot tubes that had been calibrated against the standards above referred to. A total of about 337,000,000 cu. ft. of gas was measured with a difference between the two meter records of 0.2 per cent.

25 The general formula [1] of the pitot tube and orifice is expressed by the equation

$$Q = 218.44\ Ed^2 \frac{T_s}{P_s} \sqrt{\frac{hP}{TG}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

where

E = efficiency of the tube, expressed as a decimal
d = internal diameter of the tube, in.
T_s = absolute temperature of measurement base, deg. fahr.
P_s = absolute pressure of measurement base, lb. per sq. in.
G = specific gravity of gas — air = 1.0
P = absolute static pressure of flowing gas, lb. per sq. in.
T = absolute temperature of flowing gas, deg. fahr.
h = velocity head of flowing gas, in in. of water

[1] See Appendix No. 2.

Assuming standard conditions of storage temperature and pressure, namely, $P_s = 14.65$ lb. per sq. in. absolute, and $T_s = 60 + 460 = 520$ deg. absolute, equation [7] becomes

$$Q = 7753\ Ed^2 \sqrt{\frac{hP}{TG}} \quad \ldots\ldots [8]$$

26 For any particular installation the gravity of the gas is once determined, and, with an assumed value for flowing temperature, is included with E and d to form a coefficient, making the formula read

$$Q = C\sqrt{hP} \quad \ldots\ldots [9]$$

where

$$C = 218.44\ Ed^2 \frac{T_s}{P_s\sqrt{TG}} \quad \ldots\ldots [10]$$

27 If the flowing temperature is other than the value assumed in making up the coefficient, the flow is corrected by multiplying by the factor $\sqrt{\frac{T_a}{T}}$, where T_a is the assumed flowing temperature.

28 The value of E is remarkably constant for tubes ranging in size from 2 in. to 5 in., as will be seen from the following tabulation, taken from the standard tubes already referred to:

Actual Diameter of tube, d	Efficiency, E
3.0625	0.8534
3.96875	0.8515
5.0469	0.8541
Mean	0.8530

29 Assuming the average value of E to be 0.8530, the coefficient of any pitot tube constructed as above described will be

$$C = 381.7d^2 \quad \ldots\ldots [11]$$

based on a storage pressure of 14.65 lb. absolute, storage temperature of 60 deg. fahr., flowing temperature of 40 deg. fahr., and specific gravity of 0.600.

30 In using the pitot tube it is essential that a straight run of at least 30 ft. of pipe of the same diameter as the tube be installed ahead of it, in order to avoid errors due to eddies in the flow. It is also necessary to have absolutely no leaks in the pressure lines leading from the tube to the differential gage.

31 In practice the U-gage is modified by making one leg to consist of a cast-iron water pot having a cross-sectional area many times

greater than that of the glass tube comprising the other leg of the U. Thus the height of the water level in the pot does not change appreciably for a change in the glass tube from zero to full scale reading of about 28 in. The pot is provided with a gage glass to show the level of the water in it, to which the zero of the adjustable scale is set, thus necessitating the reading of but one leg of the U-gage, that comprised by the long gage glass.

32 The scales used in the tubes described are graduated to 28 in., and are used under all conditions of pressure. If the static pressure of the flowing gas is equal to that of the atmosphere, the full scale water reading corresponds to a velocity in the pipe of over 22,000 ft. per minute. As the static pressure increases, the velocity equivalent becomes smaller and at 100 lb. gage is about 8000 ft. per minute. Thus, the higher the pressure under which the tube operates, the greater will be the range of water pressure permissible to use.

33 It is not considered advisable to use the pitot tube for flows giving a differential pressure less than 2 in. of water, owing to the high percentage of error resulting from any personal error in reading the gage at such low readings. With a maximum scale reading of 28 in. this means that under atmospheric pressure conditions the ratio of maximum to minimum quantities of gas susceptible of accurate measurement by the pitot tube is about 3.75 to 1. Assuming the same permissible velocity (22,400 ft. per minute) under a pressure of 100 lb. gage, the maximum water scale reading could be increased to more than 200 in., giving the tube a range of 10 to 1. The author is now conducting experiments along this line, but as yet has not progressed far enough with the work to state authoritatively to what extent this range may be extended without affecting the constancy of the coefficient.

34 At present the usual method of obtaining the required range of capacity in a pitot tube measuring station is to install the tubes in batteries so that more or less of them may be cut in or out of service, as conditions demand, for obtaining accurate readings.

35 In accordance with Torricelli's theorem, the velocity of issue of a fluid from an orifice follows the law of falling bodies, and the formula deduced therefrom expressing the quantity of flow is identical with that of the pitot tube. The vena contracta of the jet, however, causes the coefficient of flow (termed "efficiency" herein) to have a lower value than the corresponding factor in the pitot tube formula. For differential pressures greater than about 4 in. of water

it is sensibly constant, and thus the orifice furnishes a simple means of gas measurement that compares favorably with the pitot tube.

36 Instead of a water column, requiring personal observation at regular stated intervals, a recording differential gage may be used, constructed with the working parts enclosed in a casing to which the static pressure is applied, while the dynamic pressure is applied to the interior of the gage. Such an instrument has been used with success in connection with an ordinary recording pressure gage for determining the static pressure, thus obviating the necessity of having an attendant constantly on duty to make observations.

37 The simplicity of the pitot tube and orifice, both in manipulation and computation, and the high degree of accuracy attainable with them, render them invaluable in the measurement of natural gas in large volumes and at high pressures. The tube has been in commercial use for many years, with highly satisfactory results. The orifice, although one of the oldest known measuring devices, has not until recently been used extensively in the measurement of gas under high pressure closed flow conditions. It is being rapidly developed, however, and bids fair to come into extensive use as a simple, accurate gas-measuring device.

38 The only disadvantage in the use of these two instruments is the necessity for the computation of results. Their largest field has consequently been in the measurement of sufficient quantities of gas to warrant the maintenance of stations with observers constantly in attendance. The adoption of the recording gages, however, has greatly extended their usefulness and has considerably simplified the problem of the accurate measurement of gas under pressure.

APPENDIX No. 1

39 The general formula of the venturi meter is derived on the assumption that the change in state of the gas between entrance and throat of the meter takes place adiabatically. Hence the drop in potential energy, as evidenced by the respective pressures is equal to the increase in kinetic energy indicated by the respective velocities at entrance and throat.

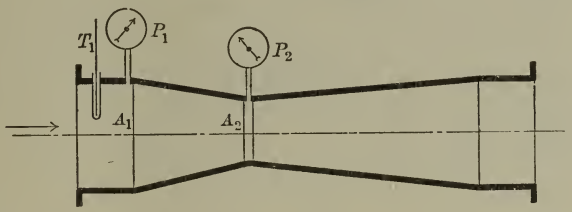

FIG. 1 DIAGRAM OF VENTURI METER

40 In Fig. 1, which diagrammatically represents a venturi meter, let

A_1 = area of cross-section at entrance, sq. ft.
A_2 = area of cross-section at throat, sq. ft.
P_1 = absolute pressure, lb. per sq. in. at entrance
P_2 = absolute pressure, lb. per sq. in. at throat
v_1 = absolute velocity at entrance, ft. per second
v_2 = absolute velocity at throat, ft. per second
W = weight of gas passing per second
δ_1 = density of gas at entrance, lb. per cu. ft.
δ_2 = density of gas at throat, lb. per cu. ft.
T_1 = absolute temperature of gas at entrance
T_2 = absolute temperature of gas at throat
T_s = absolute temperature of storage base, deg. fahr.
P_s = absolute pressure of storage base, lb. per. sq. in.
G = specific gravity of gas, air = 1.0
T_o = absolute temperature of 32 deg. fahr. = 492
P_o = absolute pressure of atmosphere, lb. per sq. in.
δ_w = weight of water per cu. ft. at 60 deg. fahr. = 62.37 lb.
δ_a = density of air at T_o and P_o
δ_{a1} = density of air at T_1 and P_1

Kinetic energy of gas at entrance $= \dfrac{Wv_1^2}{2g}$

Kinetic energy of gas at throat $= \frac{Wv_2^2}{2g}$

Increase in kinetic energy is

$$E_k = \frac{Wv_2^2}{2g} - \frac{Wv_1^2}{2g} = \frac{W}{2g}(v_2^2 - v_1^2) \quad \ldots\ldots [12]$$

41 The work done on one pound of gas in expanding adiabatically from P_1 to P_2 is

$$\frac{144\ (P_1V_1 - P_2V_2)\ n}{n-1}$$

and the change in potential energy in W pounds of gas per second is

$$E_p = \frac{144\ W\ (P_1V_1 - P_2V_2)\ n}{n-1} \quad \ldots\ldots [13]$$

42 Hence

$$\frac{W}{2g}(v_2^2 - v_1^2) = \frac{144\ Wn}{n-1}(P_1V_1 - P_2V_2) = \frac{144\ Wn}{n-1}\left(\frac{P_1}{\delta_1} - \frac{P_2}{\delta_2}\right)$$

$$v_2^2 - v_1^2 = \frac{288\ gn}{n-1}\left(\frac{P_1}{\delta_1} - \frac{P_2}{\delta_2}\right) \quad \ldots\ldots [14]$$

But $v_1A_1\delta_1 = v_2A_2\delta_2$ and $P_1\delta_2^n = P_2\delta_1^n$ or $\frac{\delta_2}{\delta_1} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$. Hence

$$v_1 = v_2\frac{A_2\delta_2}{A_1\delta_1} = v_2\frac{A_2}{A_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

$$v_2^2 - v_1^2 = v_2^2 - v_2^2\left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} = \frac{288\ gn}{n-1}\left(\frac{P_1}{\delta_1} - \frac{P_2}{\delta_2}\right)$$

$$v_2^2 = \frac{\frac{288\ gn}{n-1}\left(\frac{P_1}{\delta_1} - \frac{P_2}{\delta_2}\right)}{1 - \left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}} \quad \ldots\ldots [15]$$

43 But $\frac{P_1}{\delta_{a1}T_1} = \frac{P_o}{\delta_a T_o}$ and $\delta_{a1} = \delta_a\frac{P_1T_o}{P_oT_1}$

$$\delta_1 = G\delta_{a1} = G\delta_a\frac{P_1T_o}{P_oT_1}$$

Also

$$\frac{P_1}{\delta_1^n} = \frac{P_2}{\delta_2^n} \quad \therefore \delta_2 = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\delta_1 = G\delta_a\frac{P_1T_o}{P_oT_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

Hence

$$\frac{P_1}{\delta_1} - \frac{P_2}{\delta_2} = \frac{P_1}{G\delta_a\frac{P_1T_o}{P_oT_1}} - \frac{P_2}{G\delta_a\frac{P_1T_o}{P_oT_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}} = \frac{P_oT_1}{G\delta_aT_o}\left\{1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\} \quad \ldots\ldots [16]$$

Substituting in [15], and extracting the square root

$$v_2 = \sqrt{\frac{\frac{288\,gn}{n-1}\,\frac{P_o T_1}{G\delta_a T_o}\left\{1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\}}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}}} \quad \text{.....[17]}$$

44 The flow, in cubic feet per hour, based on P_s and T_s is

$$Q = 3600\, A_2 v_2 \frac{P_2}{P_s}\frac{T_s}{T_2}$$

But

$$\frac{P_2}{T_2} = \frac{P_1\delta_2}{T_1\delta_1} = \frac{P_1}{T_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

Hence

$$Q = 3600\, A_2 \frac{T_s}{P_s}\frac{P_1}{T_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} v_2$$

$$Q = 210{,}840\, A_2 \frac{T_s}{P_s}\sqrt{\frac{n}{(n-1)\,G}}\,\frac{P_1}{\sqrt{T_1}}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \sqrt{\frac{1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}}} \quad \text{.....[18}$$

APPENDIX No. 2

45 The general formula for the pitot tube and the orifice is derived from the law of falling bodies. Let

T = absolute temperature of flowing gas, deg. fahr.
P = absolute pressure of flowing gas, lb. per sq. in.
δ = weight per cu. ft. of flowing gas, at P and T
G = specific gravity of flowing gas, air = 1.0
v = actual velocity of flowing gas, ft. per second
h_f = height in ft. of homogeneous column of gas at P and T producing v
h = corresponding height of water column in in.
δ_w = weight per cu. ft. of water = 62.37 lb. at 60 deg. fahr.
P_s = absolute storage pressure base, lb. per sq. in.
T_s = absolute storage temperature base, deg. fahr.
δ_a = weight per cu. ft. air at 32 deg. and 14.7 lb. = 0.08073 lb.
d = actual inside diameter of pipe or orifice, in.
E = efficiency of pitot tube or orifice
Q = flow in cu. ft. per hour at P_s and T_s

Then

$$v = \sqrt{2gh_f} = \sqrt{2g\frac{h}{12}\frac{\delta_w}{\delta}}$$

But

$$\delta = \delta_a G \frac{P}{14.7}\frac{492}{T}$$

Hence

$$v = \sqrt{2g\frac{h}{12}\frac{\delta_w}{\delta_a}\frac{14.7}{492}\frac{T}{PG}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [19]$$

$$Q = 3600\frac{\pi d^2}{4 \times 144} v \frac{P}{P_s}\frac{T_s}{T}E$$

$$Q = 218.44\, Ed^2 \frac{T_s}{P_s}\sqrt{\frac{hP}{TG}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [20]$$

DISCUSSION

C. J. Bacon. The principal objection Mr. Weymouth has to the venturi meter is its complicated formula. In view of the amount of preliminary experimental work necessary to establish the coefficient of the pitot tube independently for each condition of velocity, pipe diameter and roughness of interior surface, under which it is to be used, it seems to me that the venturi meter is by far the more generally applicable, since no preliminary work of the above nature is necessary. The coefficient for gas has been determined by various investigators for tubes from 2 in. to 4 and 5 ft. in diameter and has been found to be 98 per cent for the large sizes at the most practicable velocities. By plotting to a wide scale that portion of the formula involving the ratios of areas, pressures and specific heats, the computation of results becomes at least as simple as for the pitot tube. In the usual applications a variation of one per cent in the ratio of specific heat introduces an error of only 0.02 of 1 per cent. In my work a ratio of areas of one to nine is generally maintained. The loss of line pressure through venturi gas meters having this ratio of areas has been shown to be 1/9 the differential pressure, therefore not of great importance. The various types of recording instruments on the market make possible continuous records of all data except density. I have had in use for a year or more some combined indicating and recording instruments graduated to show the actual volume of gas passing per minute. One of them is connected to a 54-in. diameter venturi meter located 600 ft. away, and in spite of the distance for transmission of a slight pressure difference it has been found very sensitive and accurate, moreover requiring almost no attention.

C. C. Thomas said that of course it required electric energy to operate the electric gas meter, about 1 or 1½ cents worth per 100,000 cu. ft. of gas measured. It required about 1 kw-hr. per 100,000 cu. ft. of gas. The necessity of increasing the velocity of gas in order to measure it by means of pitot tubes also required energy. In order to produce a readable height of water column (it should not be less than 4 in.) the velocity of gas was increased at the pitot tubes by an amount requiring about 1 per cent more energy to be expended in the pumps than was required with the electric meter installation. This was the case also when the measurement was made by means of orifices, as it was necessary

to produce a fall of pressure of about 1 lb. per sq. in. from one side to the other of the orifice. This reduced the suction pressure on the pumps.

Deposit on pitot tubes had a serious effect on accuracy though it was possible to avoid this to some extent by using screens. The pitot tubes were standardized in place, but it was very difficult to check them afterwards. An electric meter could be checked for accuracy at any time simply by throwing in a switch and noting whether or not the system came to a balance. No computation of results or reference to charts was required and no straight length of approach was necessary.

E. A. Hitchcock (written). In 1908 with Mr. Eugene Coste of Toronto I had occasion to standardize with natural gas a set of three pitot tubes, 2, 3 and 4 in., similar to those described by Mr. Weymouth in Par. 27. A total of 34 runs were made under a variety of velocities and pressures, the quantity of gas flowing being determined by means of a gas holder having a capacity per foot of rise of 5246.5 cu. ft.

In order to reduce to a minimum the effect of temperature on the gas holder all the experimental work was carried on during the night and to obtain parallel flow there was laid a straight run of pipe for 92 ft. leading to the brass pitot tubes.

For gas having a flowing temperature of 40 deg. fahr. with a storage temperature of 50 deg. fahr. and a pressure of 14.7 lb. the constants K were found to be 1511 for the 2-in. tube, 3384 for the 3-in. tube and 5734 for the 4-in. tube where

$$Q \text{ (cu. ft. per hr.)} = K \sqrt{\frac{h\,P500}{T}} \quad (0.668 = \text{sp. gr. gas})$$

The actual diameters of the tubes calibrated were 2.08 in., 3.07 in. and 3.98 in. respectively. It was found, as stated by Mr. Weymouth, that for each tube throughout a wide range of velocity head readings, the coefficient was almost constant.

Taking the constants as given above, solving for Q for any condition, making the proper substitution in equation [7] and solving for E, I find that E is not constant for the different tubes but decreases with the diameter of the tube:

Actual diameter of tube d, in.	Efficiency E
2.08	0.843
3.07	0.866
3.98	0.874

This is explained in that no correction is made for the area taken up by the dynamic tip in the brass tube. For the above cases this tip had an outside diameter of ½ in. and therefore E will decrease until the actual diameter of the pipe is ½ in. when E would become zero because Q would be zero in that the tube would be plugged with the dynamic tip. For this reason it would seem that d should be taken in actual equivalent diameter.

Mr. Weymouth states that it is not advisable to use the pitot tube for flows giving a differential pressure of less than 2 in. water. Judging by my experience with the type of apparatus he has described, I can readily see why this may be advisable. The reading scale is graduated in 1-10 in. and the average observer will read to the nearest 1-10 in.; therefore with a 2 in. head of water, there might be an error with one reading of 2.2 per cent. Also there seems to be considerable vibration to the water meniscus so that it is difficult to read to less than 1-10 in.

To accept a 2-in. minimum would as a general rule eliminate the use of the pitot tube for measuring natural gas in a great many cases, for of the many hundreds of readings I have taken during the past 13 years, in the natural gas field on lines from wells to mains and mains to stations the velocities which gave 2 in. or more of water were the exception.

The form of instrument used was a traversing dynamic tube made of 1.4-in. brass pipe, a stationary static tube of the same size and material and the form of manometer brought out by Prof. S. W. Robinson. With this form of apparatus and using alcohol in the manometer there is no difficulty in reading to 1-100 in. which, with a velocity head of only ½ in. and an error in reading of 1-100 in., would mean an error in the quantity of gas flowing of only 1.4 per cent and this probable error is reduced by taking many independent readings.

The greatest advantage of the pitot tube in the measurement of natural gas is its great simplicity and ease of application. Several determinations can be made during the same day on different lines in the same field in that it should not take more than one hour to connect the instrument to the line after the gas pressure has been reduced to 20 lb. or less.

The Author. In referring to the preliminary work necessary to establish the coefficient of the pitot tube Mr. Bacon doubtless had in mind the form of tube intended to be used within an ordinary pipe. In this form the pitot tube is more of a portable

instrument than one adapted for permanent installation, and should not be compared to the venturi meter. For such comparison the form of tube described in Par. 22 must be chosen, inasmuch as its field of usefulness is the same as that of the venturi meter. The total amount of preliminary work necessary with this type of tube is merely its calibration against a standard tube or other accurate measuring device and involves from one to four hours' time, which seems to the author to be about as little as one could expect to be spent upon any instrument of the importance of those considered. Much labor and painstaking care were applied to the derivation of the coefficients of the standard tubes mentioned in Par. 23, but these coefficients having once been determined, it is a simple process to ascertain from them the coefficients of the other tubes calibrated against them. The standard tubes are preserved in a testing station used solely for the calibration of other tubes and are in service only during the progress of such calibration work.

The computation of the quantity of gas passing through pitot tubes in commercial use is extremely simple. From a book of tabulated values of $\sqrt{h\,(p + 14.4)}$ corresponding to all values of h from 0 to 28 in. of water, by tenths, and of p from 0 to 500 lb. gage, by pounds, the value of $\sqrt{h\,(p + 14.4)}$ corresponding to observed readings of h and p is read off and written on a form every 15 minutes. At the end of the day these "extensions" or values of $\sqrt{hP}$ are added and the sum multiplied by the quarter-hour coefficient of the tube used, giving the total quantity of gas passed. The method of computation of venturi meter observations as outlined by Mr. Bacon does simplify the use of the formula of Par. 12, but can hardly be said to be as simple as the pitot tube computation, as it would require first, the computation of $\frac{P_2}{P_1}$ from the observations of P_1 and $P_1 - P_2$. Then the corresponding value of the radical must be obtained from the chart. This value must then be multiplied by the value of P_1 and also by a constant, depending upon the particular venturi instrument used and the remaining factors in the formula. This computation could all be reduced to much simpler form, however, if tables were computed giving the values of $\frac{Q}{A_2}$ for all possible combinations of

P_1 and P_2. Such tables would then be applicable to all venturi meters with the same ratio of entrance area to throat area.

The cost of measuring gas by pitot tube as influenced by the drop in pressure can be made small by properly designing the station. In the case assumed by Professor Thomas, the pressure drop of 28 in. or about 1 lb. would be equivalent to 1.2 hp-hr. per 100,000 cu. ft. per hr., assuming the gas to be pumped from 80 to 350 lb. gage pressure. By using a 6-in. tube instead of a 5-in., the drop in pressure would be reduced about 50 per cent, measuring the same quantity of gas, and the cost would thus be cut in half.

It is true that if pitot tubes should be filled with deposit their coefficients, which vary as d^2, would be seriously affected. As a matter of fact, however, such deposits do not form, due to the high velocity of the gas in the polished tube. Sometimes the tips will become clogged with dirt, but this instantly becomes apparent on the gages and the tips can then be "blown out" or removed from the tubes and cleaned.

All pitot tube stations should be installed with provision for inserting a portable standard tube for calibrating the permanent tubes. It is then a simple matter to standardize all the tubes in a station as frequently as is necessary.

The coefficients of the tubes referred to in Par. 28 were determined in the same manner as those described by Professor Hitchcock and the author is unable to understand the variation in the values of E as shown by him. Even though the tip were assumed to be so large as to fill the pipe it would not mean that E would fall to zero, for while Q would then be zero, h would also be zero, and E could remain constant.

The limits mentioned in Par. 33 apply only to installations of the type described, and are a result of the simple form of U-gage scale usually used with such permanent stations. When a properly designed scale is used for the accurate reading of small differential pressures, the range of the pitot tube is increased to give accurate measurements over extremely wide limits, for the fundamental principle of the tube itself holds good irrespective of the velocity, and the question resolves itself into one of the accurate measurement of small differences in pressure. This is especially true in portable tubes or tips. The author makes use of such a form in field work, especially in the measurement of the capacity of wells feeding into main lines, and obtains

very accurate results with small differentials by using an inclined U-gage.

Where trouble is experienced from vibration of the water meniscus, it can be overcome by inserting in each of the pressure lines a diaphragm with a hole in it about 1/32 in. in diameter. If the installation is permanent the same object may be attained by inserting in each of the pressure lines a joint of 6-in. pipe, capped on both ends, for acting as a reservoir or cushion to dampen the vibrations and give a true average of the pressures.

No. 1377

AXIOMS CONCERNING MANUFACTURING COSTS

By Henry R. Towne, New York

Member of the Society

Three factors enter into the cost of each and every article of manufacture; namely, *materials*, *labor* and *expenses*. These constitute a tripod, a three-legged stool, which cannot stand if one of these legs be omitted. They may, and do, vary in dimension, but all three are invariably present, and a "cost" which omits any one of them is incomplete and fallacious. The formula is

$$L+M+E=C$$

in which L represents labor, M materials, E expenses and C cost. In this primary division the item "labor" includes all labor entering directly into the product, the item "material" all material entering directly into the product, and the item "expenses" (often called overhead charges, or simply overheads) all other labor, material and expenditures of every kind whatsoever.

Axiom 1 *Every cost includes three fundamental factors: labor, material, expenses.*

2 In most cases, however, the expenses, or overheads, divide naturally into two groups:

a Manufacturing expenses, those incident to the operation of the factory or mill; that is, those incurred in utilizing productive labor and material, and in bringing the product up to the point where it is ready to be sold.

b Commercial expenses, those incident to the commercial department of the business, including administration, salesmen, advertising, office expenses, etc.; that is, those incurred in distributing and selling the finished product.

It is highly expedient that these two should be segregated, so that each may be studied separately.

Presented at the Annual Meeting 1912, of The American Society of Mechanical Engineers.

Axiom 2 The expense factor should be split into two parts: manufacturing, commercial.

3 Letting the symbol *Me* represent the former, and the symbol *Ce* the latter,

$$L+M+Me+Ce=C$$

But a more convenient and indicative form of presenting these elementary facts, one which the writer has used for many years, is the following:

L = productive labor

$\underline{M}$ = productive material

$\overline{PC}$ = *prime cost*

Me = manufacturing expenses

$\overline{SC}$ = *shop cost*

$\underline{Ce}$ = commercial expenses

$\overline{AC}$ = *actual cost*

4 If preferred, the foregoing facts may be expressed by the following formulae:

$L+M=PC$ or *prime cost*

$PC+Me=SC$ or *shop cost*

$SC+Ce=AC$ or *actual cost*

Axiom 3 A manufacturing cost has three phases: prime cost, shop cost, actual cost.

5 On the appreciation and intelligent use of these facts hang all the laws of good business and the profits, for no business can long be operated successfully without a correct knowledge of costs, nor can that be had without a clear grasp of fundamental principles. The competitor most to be feared, while he lasts, is one who does not know his costs, nor understand how to obtain them.

Axiom 4 Accurate cost information is vital to good management.

6 Simple as are these elementary principles, their correct application in each given case is difficult, and calls for great care and intelligence. To draw correctly the line between productive and non-productive labor and material, through each of the successive stages of a productive industry, requires the combined skill of the expert manufacturer and the expert industrial accountant; the former knowing accurately all the details of the manufacturing or productive processes, and the latter knowing equally the proper methods of combining and using the recorded facts to yield the desired information. For example, what constitutes productive

labor? In the case of a machinist operating a lathe, clearly it includes his wages while his lathe is turning out product and also while it is standing still during the time he is dressing the tool to do the work properly. But if the tool-dressing is done for him, as it is under good modern practice, how shall the time and wages of the tool-dresser be classified? So as to productive material, shall the tool, the file, the waste, the oil, which are consumed or used up in making the product be classified as productive or as non-productive material? The answers to these questions depend on the surrounding facts in each case, and are as varied as the cases are infinite in number and variety. The writer is not attempting here to answer such questions, but merely to point out and emphasize underlying principles. This much is clear, that every individual item of expenditure, large and small, must ultimately classify under one of the three great heads above referred to, labor, materials or expenses, and that profit, or loss, is the difference between actual cost and the net price realized.

Axiom 5 Accurate costs imply the correct classification of every expenditure.

7 The distribution of actual costs among these heads, or, preferably, into the four groups or divisions indicated above, varies widely in different industries and with different products. This is illustrated by the following table, relating to four distinct lines of actual product, in which the several elements have been reduced to terms of the actual cost of the product when finished and sold.

Nos.	1	2	3	4
L = productive labor.....	28	17	29	19
M = productive material.....	38	33	25	37
PC = *prime cost*.....	66	50	54	56
Me = manufacturing expenses..	24	20	28	22
SC = *shop cost*.....	90	70	82	78
Ce = commercial expenses.....	10	30	18	22
AC = *actual cost*.....	100	100	100	100
Total expenses (that is, $Me+Ce$) =	34	50	46	44

8 The figures in the above table illustrate the hopeless state of mind of those manufacturers, some of whom still survive, who delude themselves by the belief that the sum of labor and materials (prime cost) represents the actual cost of the product, and that the difference between that and the selling price is profit. They show, on the contrary, that, in the four examples to which the figures relate, the

prime cost constitutes only from one-half to two-thirds of the actual cost, and that the expenses, or overheads, incident to conducting the business and marketing the product, contribute from one-third to one-half of the total or actual cost. It seems probable that, if the facts concerning all manufacturing industries could be ascertained and averaged, the "three-legged stool" would be found to stand nearly level, its three legs being approximately of equal length, although differing widely in individual cases.

9 All cost accounting should aim to segregate charges wherever this can be done accurately. Thus the major part of the items constituting productive labor and material can and should be charged directly to their respective accounts, L and M.

Axiom 6 Every productive expenditure should be charged directly to its proper account.

10 All other items, however, which cannot be so segregated must be aggregated into one or several groups and their total apportioned among the proper accounts on some carefully determined but necessarily arbitrary basis.

Axiom 7 All non-productive expenditures should be properly grouped for final distribution.

11 Manufacturing expenses may be apportioned as a ratio or percentage of labor L, of material M, or of labor and material $L+M$. The usual bases are either L or $L+M$. The writer believes that in most cases the closest conformity to actual facts will be attained by distributing manufacturing expenses in the ratio of productive labor, hand and machine, because usually the volume of indirect expenses of works operation will be far more influenced by the pay-roll, that is, by the number and kinds of employees, than by the materials bill, that is, the amount paid for the material of production. Moreover, the former is relatively stable, while the latter fluctuates with market changes. Therefore it is advisable that Me should be a function of L, that manufacturing expenses should be apportioned as a percentage of productive labor, although in some cases they may properly be apportioned per machine, or per unit of floor space.

Axiom 8 The normal basis for distributing manufacturing expenses is productive labor.

12 Commercial expenses may also be apportioned as a percentage of L, of M, or of $L+M$, and frequently are, but more properly they should be apportioned on the basis of shop costs, $L+M+Me$. The reasons for this are conclusive. Production and selling are two sepa-

rate and distinct processes. The former brings the product to the point where it is completed and ready for sale; the latter then takes it over and effects the sale. The expenses incurred in each process are for its use only, and have no natural relation to the needs and uses of the other. To illustrate this, take the case of a manufacturer of cotton cloth who sells his entire product through a commission house or broker. Clearly his whole commercial expense account is covered by the commission he pays to his selling agent, and this bears a definite ratio to his shop cost, that is, to the cost of his product ready for sale. Now, take the case of another manufacturer of cotton cloth, who maintains his own selling organization and through it distributes his product. Clearly his commercial expenses offset the commission paid by his competitor, and equally bear a definite ratio to his shop cost. Both are most accurately stated and apportioned as a percentage of the shop cost, the cost of the product ready for sale. Therefore commercial expenses *Ce* should be distributed as a percentage of shop cost, $L+M+Me$.

Axiom 9 The normal basis for distributing commercial expenses is shop cost.

13 When the product is simple and homogeneous, for example, such as pig iron or cotton cloth, one account may suffice for all manufacturing expenses and one other for all commercial expenses, but when it is diverse or complex each of these should be subdivided into one for each department or for each distinct class of product. In effect such a business is an aggregate of several businesses, some of which may yield better results than others, or may fluctuate more widely, and a proper accounting system should show the results of each subdivision or department separately, as well as the combined result of all. Hence arises in many cases great complexity in cost accounting, and corresponding need and opportunity for the skilled industrial accountant.

Axiom 10 An accounting system should show results both by departments and by totals.

14 In some cases the entire product consists of a single staple article, or group of articles, for which there is a constant demand and, at some price, a sure sale, such as pig iron, window glass, cotton cloth, etc. In other cases the product must conform to the specifications of the customer, and therefore cannot be made up in advance of orders, as in shipbuilding, carbuilding and the construction of buildings. The former is commonly designated as a

stock product, and the latter as contract work. The difference between these may be expressed as follows, viz.:

A stock product is one which is made first and sold afterwards.

A contract product is one which is sold first and made afterwards.

Cost accounting is usually more complex and difficult in the case of contract work than in that of a staple or stock product.

Axiom 11 A contract product may require a more complex accounting system than a stock product for the accurate determination of costs.

15 The expenses of general administration overlap the manufacturing and the commercial divisions of an industrial business. Many items can and should be definitely charged to one or the other. Others may arbitrarily be apportioned between them; as, for example, the salary of an official who devotes say 70 per cent of his time to one and 30 per cent to the other. All others must be aggregated into groups for distribution by the methods adopted, as above, for distributing such expenses; as, for example, by percentages of productive labor or of shop costs. Expenditures of this kind are infinite in size, kind and number, and call for great skill and good judgment in their classification, which should be determined in advance by a clearly defined code, not left for haphazard decision by subordinates. Such a code, based on intimate knowledge of the business, on a clear perception of the information the code is designed to yield, and on sound accounting principles, is an indispensable prerequisite to the accurate determination of costs, and equally to the intelligent conduct of any manufacturing business.

Axiom 12 An accounting system should be embodied in a code of instructions, for the guidance of those responsible for its operation.

16 For best convenience a code should provide symbols to represent the various accounts and their many combinations. For this purpose the writer for many years has used a system of letters and numbers which possesses great convenience. Letters are used to designate important departments and accounts, the significance of each letter depending on its place, as in decimal notation, in the symbol. Thus, the first letter may indicate a department, the second a subdivision of it, and the third a room or smaller unit. Stated numbers are used to indicate accounts relating to expenses of the

various kinds or groups. Such a symbol is shown by the following example, viz.:

BAC 10

in which *B* represents the department, or the class of product, against which the item is to be charged; *A* the shop in which the work is done; *C* the job, or machine, by which it is done; and 10 the kind of expense to which the charge relates, such as repairs of the machine, foreman's wages, etc. These symbols, and an explanation of their meaning and use are printed in a small book of pocket size, copies of which are furnished to all concerned. In this way a correct classification of every charge is made at the time of original entry, after which tabulation and aggregation of original charges follow automatically in accordance with the predetermined plan.

Axiom 13 Symbols are better than titles for recording charges in an extensive accounting system.

17 In any business certain expenses or losses occur from time to time which are unusual or abnormal. These may be termed "extraordinary expenses," and require special consideration. As examples of these may be cited a serious loss by fire, a curtailment of product by a strike, an abnormal loss through bad debts, an increase or decrease in value of land, etc. The loss, or profit, thus arising must of course be covered into the treasury, but this may better be done through a debit or credit to the surplus account than through a charge to the profit and loss account of the current year, for the latter plan would distort the statistical record of the year by including in it items not common to normal years. The best plan is to charge to the account of each year only the items which are normal, and to charge those which are abnormal to the surplus account. The proper purpose of the annual account is twofold, (*a*) to show the results of the year's operations, and (*b*) to contrast these results with those of preceding and succeeding years. On the other hand, all extraordinary gains or losses must be accounted for, and this may best be done through the surplus account. In this way both purposes are accomplished.

Axiom 14 Extraordinary gains or losses, in order not to distort the statistical value of the annual profit and loss record, should be covered into the surplus account between the closing of the books for the old year and the opening of the books for the new year.

18 Interest on borrowed capital is a distribution of profits, not

an expense, although often erroneously treated as the latter. To illustrate this suppose the case of two manufacturers, *A* and *B*, each having $200,000 invested in his business and each realizing 10 per cent, or $20,000, net profit available for dividends on a year's business, all of *A*'s capital being contributed in cash, while *B* has only $100,000 of cash capital, and another $100,000 of borrowed capital, on which he pays 5 per cent interest. At the close of the year *A* is in position to pay $20,000 in dividends to his stockholders, a 10 per cent return on their investment, but *B*, after paying $5000 as interest, is in position to pay $15,000 in dividends to his stockholders, a 15 per cent return on their investment. Evidently the actual profits from the operations of the year are the same in each case, only the ownership of the capital invested and the distribution of the profits being different. The accounting system should show the actual profit realized, regardless of its distribution to the owners of the capital invested in the business. On the other hand it is expedient that interest on temporary loans, and on time purchases if availed of, rebates and discounts of customers' notes, should be treated as current expenses, normal to the conduct of the business. In like manner discounts earned by cash payments should be treated collectively as part of the current earnings of the year, or else be covered into the net costs of purchases.

Axiom 15 Interest on borrowed capital should not be treated as an operating expense, but should be charged direct to the profit and loss account of the year.

19 Interest on all capital invested in a business may or may not be deducted before stating the final profits of the year. Here no principle is involved, but merely convention or individual preference. Usage, however, has practically determined that it shall not be deducted; that the final, or net, profit should indicate the return on capital, the amount which capital has earned. Stated thus, it can readily be compared with what the same capital would earn if invested otherwise; as, for example, in government or railroad bonds, in mortgages, etc. If interest is deducted at all, as is done, for example, under some profit-sharing plans before allotting anything to the beneficiaries of the plan, it should be computed on the total capital invested in the business, including therein the surplus account; that is, surplus profits of previous years retained in the business and invested in plant or merchandise.

Axiom 16 Final profits properly signify the amount earned by

the capital invested. If interest on capital is deducted this fact should be stated, and interest should be computed on the total capital employed.

20 Where a business is divided into several or many departments it is very desirable that the accounting system should show the profits or earnings of each of them separately, and this is usually feasible, except as to annual depreciations and as to interest charges. In some cases either or both of these items can accurately be distributed among the several departments, and if so, they should be so treated. Where they cannot be so distributed they should be deducted in a lump from the sum of departmental profits, and in this case it becomes convenient to adopt terms to designate clearly the profit account at its various stages. For this purpose the writer has found the following terms satisfactory:

Gross profits: the aggregate profits of all departments, prior to deducting depreciations and interest.

Net earnings: the gross profits after deducting depreciations.

Net profits: the net earnings after deducting interest on borrowed capital.

21 In comparing the results realized in two or more comparable concerns or businesses it is essential to contrast profits at the same stage in each case, and to employ terms which are mutually understood as to their precise meanings. No standard as to these terms has yet been established. The proper basis of comparison usually is that indicated above by the term net earnings, which eliminates the variations due to the employment or non-employment of borrowed capital.

Axiom 17 Terms used to designate profits should indicate clearly the stage of profits to which they refer, and should be mutually understood.

22 Inventory valuations are an important factor in determining profits. Usually an actual inventory is taken only once a year. The merchandise inventory includes raw materials, stock in process, finished goods, and general supplies. A standard basis of valuation for each of these groups should be adopted and maintained from year to year. Raw materials, such as pig iron, raw sugar, baled cotton, ingot copper, etc., are often subject to wide fluctuations in market values or costs, and the question thus arises as to the proper inventory valuation of them, whether at cost, at market value at date of inventory, or on some arbitrary basis. If the effect of such

fluctuations is negligible, that is, is small in ratio to the annual total of the account, either of the first two methods above stated may be used. If the fluctuations are large, however, either in range or in their effect on the annual total, that is, if they materially influence the profit and loss account of the year, some arbitrary plan of accounting for them should be adopted. In devising this, the twofold purpose of the annual account, the operative and the statistical, should be kept in view. If the effect of the fluctuations is moderate in its ratio to the annual account a sound method consists in taking the mean, either of market prices or of actual purchase prices for say three or five years, as the basis for inventory valuations, and also for use in the compilation of costs, thus conforming to the average trend of market values but avoiding frequent and temporary changes. If, however, the effect of these fluctuations is serious or vital in determining the results of the business, a new factor is brought into the accounting problem, namely that of trading or speculating on the market. In the case of a sugar refinery or a cotton mill, for example, large profits or losses may result from market changes in the price of raw sugar or of baled cotton, or from the operations of the purchasing department. Obviously such gains and losses are totally unrelated to the economy and efficiency of the productive department, and to include them in its accounting might so distort it as to destroy its usefulness and its statistical value. In such cases a separate trading account should be established, through which to ascertain the profit or loss of the year in operating on the market for the raw material, the latter being charged to the manufacturing department at a constant price, conformed from time to time to average market conditions, this price being used also for inventory and cost purposes. In other words, the results of speculation on the market, however legitimate or necessary, should be segregated from the results of the normal operations of the plant.

Axiom 18 Speculative profits and losses should be segregated from those due to the normal operations of a business.

23 The inventory valuation of stock in process, that is, of stock in a partly manufactured condition, should be such as to cover the prime cost of the material, and of the productive labor already expended upon it, plus a ratable charge for manufacturing expenses. The inventory valuation of finished stock, that is, of stock completed ready for sale, should be on the basis of shop cost, not of actual cost, because the latter includes the cost of selling, and this has not yet

been incurred. An apparent paradox, not, however, real, created is when the cost of a product is substantially reduced, because thereby the inventory value, and therefore the profit of the year, is reduced. If the inventory value at the beginning of the year were $1000, and if during the year the cost were reduced 10 per cent, obviously, if the quantity on hand at the close of the year were the same, the inventory value would be $900, thus showing a shrinkage of $100. In the following year, however, this apparent loss would be converted into an actual profit.

Axiom 19 A reduction in cost implies a corresponding reduction in inventory.

24 The annual inventory may properly include as assets certain items previously classified as expenses. One example of this kind is the premium on unexpired insurance. Another is the cost of a trade catalogue intended to serve say for five years. To charge the whole of important expenditures of this kind into the current expense account of the year in which they are incurred would tend to distort its statistical accuracy, and hence would be bad accounting. The proper treatment of such expenses is to determine the period they apply to, and to charge off a proportionate part in each month or year during that period, carrying the remainder in the inventory.

Axiom 20 Expenditures in one year which cover the requirements of several years should be distributed over the years to which they fairly apply.

25 The inventory valuation of all property other than merchandise should be on the basis of its fair value in the business as that of a going concern, which usually is the cost to replace, with due allowance for wear and tear. An annual inventory of all property, by actual enumeration and count, is indispensable to the proper conduct of any manufacturing business, and in some cases more frequent inventories of the merchandise stock are expedient. Without such annual inventories no determination of annual results is reliable or of much value.

Axiom 21 An annual inventory of all property is indispensable to accurate knowledge and to good management.

26 The question of depreciation of fixed property enters into all industrial accounting, and should be treated in connection with the inventory. In this, as in all discretionary matters of accounting, the aim should be to find and follow the median line, the mean between ultra-conservatism and radicalism. All fixed property, ex-

cepting land, depreciates and tends to become obsolete. Normal repairs and maintenance should of course be charged to current operating expenses, not added to cost or value, and these should fairly be considered in fixing the ratio of depreciation. Where a building or a machine is maintained in perfect condition obviously it depreciates more slowly than one which is neglected. A building may be so maintained as to depreciate little or not at all. The proper rate of depreciation for each class or kind of fixed property is a matter of good judgment, for which no rules can be laid down. It may be as low as 1 per cent per annum, and in exceptional cases may be as high as 20 per cent. Usually it ranges from 2½ to 10 per cent. When profits are abnormally large the allowance for depreciation may wisely be larger than when they are merely normal, but the normal allowance should be made even when no profit is realized. Under average conditions it usually ranges between 10 and 15 per cent of the annual profits. A revaluation of all fixed property by outside experts or appraisers, at intervals of five or ten years, is expedient and usually worth its cost. Abnormal increases or decreases in the value of such property, as for example an increase in the value of land, or the loss due to the demolition of an obsolete building, should be covered into the surplus account, not into the profit and loss account of the year.

Axiom 22 Valuations of fixed property should be subject to annual review and to fair depreciation.

27 Finally, the aim and object of every accounting and cost system should be to afford true and accurate information as to facts. It is based on facts; it should embody and present facts, and naught else. To exaggerate facts and to show fictitious profits and values, is no worse than to depreciate facts and to conceal true profits and values.

Axiom 23 An accounting system should present facts, without bias in any direction.

28 Accounting, in its application to general business affairs, has long been a highly developed science, but is comparatively a new one in its specialized application to modern industry, with its vast and complex development. The creation of a *Correct Science of Industrial Accounting* and costs should be the desire and aim of all who are concerned with industrial management. To accomplish this, three things at least are needed:

a A clear understanding of fundamental principles

b A definite terminology, generally understood and accepted

c A free interchange of the data of practice, whereby the adoption of sound principles may be promoted, the experience of each may be available to all, the best methods may become established, and, above all, a standard system may ultimately be created.

29 The accomplishment of these results, by affording complete and accurate knowledge of the essential facts pertaining to industrial efficiency, and to the costs of production, will tend to promote greatly and permanently the development of American industry, and to aid it in securing its full share of the markets of the world.

DISCUSSION

Arthur C. Jackson. I wish to point out a different procedure from the one suggested by the author in Axiom 9. If the interpretation of the author's statements and the arguments I wish to present for consideration can be accepted, a somewhat different result than that which has in the past been generally acknowledged correct will have to be considered.

If in Par. 12, " production and selling are two separate and distinct processes " (and I would be willing to differentiate still more decidedly between them and call them separate and distinct functions), is there any reason for distributing commercial expense upon the bases of shop costs?

The statement in Par. 11 advocating the elimination of material from the basis of distribution of non-productive or manufacturing expense on account of its varying value should lead to the elimination of this variable component from the basis of distribution of commercial or selling expenses. This leaves only two bases upon which commercial or selling expenses may be distributed, productive labor or selling prices.

Inasmuch as " producing and selling are two separate and distinct processes," does it not appear that the total selling expense should be distributed upon the basis of sales, just as the total of all expenditures incident to, and not an element of, the production or the product, are distributed upon the basis of productive labor?

Is it not often the case that the selling price of the manufactured product remains more uniform than the prices of the materials entering into its manufacture; and that the selling

price is much more affected by market and trade conditions than by variations in factory costs?

It is always the ideal of the efficient works manager to get out an ever increasing production with an improvement in quality and a reduction in costs. Likewise it is the object of every sales manager constantly to increase the volume of sales and the profit and reduce the proportionate selling expense.

Is not, therefore, the total value of sales the proper basis for the distribution for commercial or selling expenses? This is where the principle that an article can have but one cost fails and so may my argument, but I submit it for consideration because I believe it to be convenient, especially where we have to compete in export markets with foreign competition at a lower price.

If my method of determining cost by basing commercial or selling expense upon the sales instead of upon the manufacturing cost could be accepted and used it might result in less oppression of the American manufacturer by the departments of justice and internal revenue and tariff boards, as we could then *show* that goods sold in export markets at a price below the domestic actually have a lower cost than goods sold in the domestic markets at a higher price.

Augustus Smith. The title of Mr. Towne's paper, Axioms Concerning Manufacturing Costs, does not at once suggest the full scope of the paper. We want to know the final or "actual cost," as Mr. Towne puts it, to know what to charge for our product as well as to analyze and study all the various elements that go to build up the cost in order to improve and make more efficient the other departments of a business after the manufacturing department has done its best.

In addition to all this, we want to know what is our actual profit in order to pay proper dividends or to settle equitably all claims based on the cost or on a share of the profits. By a provision of the Payne-Aldrich bill passed August 5, 1909, manufacturing corporations are required to report to the Internal Revenue Bureau what their annual net profits are, and are required to pay a tax on any such profits that are in excess of $5000 per annum.

Thirty-four or more states have passed legislation authorizing a federal income tax. The difference between apparent, estimated or scientifically determined profits will soon make a difference in the taxes we have to pay as individuals.

The writer hopes that the Society will take steps to have the subject treated by Mr. Towne investigated by a committee under his guidance with a view to reporting upon a method for computing costs that will be generally accepted as standard

a For paying dividends
b For settling claims based on a share of the profits
c For paying for or receiving payment for work done on "cost" plus a percentage or "cost" plus a fixed sum
d For paying income taxes.

There is no standard practice now. There is no room for bitter disputes on any of the above calculations.

On the back of the blank which the writer's corporation filled out for the Internal Revenue Bureau are printed instructions as to how income shall be determined. This is an extract:

16. The gross income of mercantile corporations should be ascertained in the following manner: From the sum of the total sales during the year plus the amount of the inventory at the end of the year deduct the sum of the inventory at the beginning of the year plus the cost of goods and materials purchased during the year. To this difference add the income received from any other source, and the result will be the gross income to be reported under item No. 3 of the return.

17. The gross income of manufacturing corporations should be ascertained as in No. 16 except that the cost of manufacture should be included in the deduction to be made from the sum of the sales made during the year and the inventory at the end of the year.

It is only necessary to point out that in many businesses the execution of orders taken or sales made in one year may extend far into the next year or the year thereafter to show that the method of accounting suggested by the government does not in all cases show the correct income of a corporation in the sense probably intended by the act.

The method which Mr. Towne has suggested of keeping separate account of the surplus has a direct bearing on this question. For instance, a firm's receipts in one year might far exceed the disbursements, due perhaps to deferred payments coming in or to the cumulative or fortunate result of advertising or of previous soliciting or what not, and without an accepted system of surplus accounting, such a firm might be mulcted for a heavy tax to the government or might have to pay out large sums to parties interested in a "share of the profits," notwithstanding the fact

perhaps that for several years previously the firm had been apparently steadily losing money and might expect deficits again for the next few years.

The cycle of planting, cultivating and harvesting in a manufacturing business is not necessarily coincident with the calendar year, and the status of such a business cannot be shown properly at fixed yearly periods without a surplus and shortage account. If a business had to borrow money for several years before revenue began to come back, any funds so borrowed should be repaid before any profit could be declared. Just so with the surplus and shortage account.

The writer does not quite follow Mr. Towne in the matter of interest. Interest on borrowed capital which the author classes as part of the profits seems to the writer to be more conservatively considered as an expense and part of the cost. A sharp distinction can be drawn between interest and other forms of profit. Interest on borrowed capital has to be paid whether or not there is any profit in the business. The stockholders may levy an assessment or borrow more money to pay it, but pay it they must or turn over the business to a receiver.

The author cites two profitable businesses, one where *A* puts in $200,000 as capital while in the other *B* puts in $100,000 as capital and borrows $100,000 at 5 per cent. Each makes $20,000 which is 10 per cent on *A's* capital while *B* divides his $20,000 of profit into 15 per cent on his capital and 5 per cent interest on his borrowed money. But suppose each business lost $20,000. This would be 10 per cent loss on *A's* capital, but *B* would have to pay 5 per cent on his borrowed money just the same and face a loss of $25,000 on his capital or 25 per cent. The fact of the matter is the borrowed capital is not " on the table " in the sense of being exposed to the hazard of the business.

Another way of looking at borrowed money is that many businesses, especially those which the author characterizes as engaged in contract work, have no definite amount of invested capital, in his conception of the term. Such firms take as much business as they can get hold of and then raise the necessary additional capital to handle the business just as they take on extra hands to execute it, and the interest they pay for this capital must be considered an expense the same as are the wages they pay for the extra men and to be deducted before any profit is declared. The writer thinks therefore that Axiom 15 should be amended to read

Interest on Borrowed Capital should be treated as an Operating Expense.

The object of this discussion, however, is to emphasize the vital importance of Mr. Towne's paper calling as it does for agreement in the method of accounting or the creation of a correct science of industrial accounting. The writer's appreciation of the value of the paper is such that he has hesitated to express a divergent opinion concerning Axiom 15. The main thing is to work for a standard system which can be amended from time to time afterward if necessary. As the author says no standard as to what such words as "earnings" and "profits" mean has as yet been established. Scientific management without scientific accounting resembles professional billiard players competing without keeping the score.

Wm. Kent. Referring to Mr. Towne's percentage distribution of the several items of cost given in his table in Par. 7, he takes the total cost, including the commercial expenses at 100 per cent and expresses the several items as percentages of that total cost. In my experience as an accountant I have found it preferable to take the factory cost of the goods packed and in the warehouse at 100 per cent, and to express the several items as percentages of that total. By this method the table would be modified as follows:

Nos.	1	2	3	4
L = productive labor	31	24	35	24
M = productive material	42	47	30	48
PC = *prime cost*	73	71	65	72
Mc = manufacturing expenses	27	29	35	28
SC = *shop cost*	100	100	100	100
Cc = commercial expenses	11	43	22	28
AC = *actual cost*	111	143	122	128
Total expenses (that is, $Mc + Cc$) =	38	72	57	56

The factory cost is the labor, plus the material, plus the factory expense. The factory should make the goods as cheaply as it can and sell them to the selling department either at an agreed price or at some percentage of the factory cost. In a factory in which I was interested we arbitrarily fixed the price to the sales department at 50 per cent of list (we had a list price from which dis-

counts were usually given), and we said that the factory should make the goods for half the list including the factory expense, and the factory should make a profit on them at that price if it could. The selling department should buy the goods from the factory at 50 per cent from the list and sell them for what it could, and make as much profit as it could. The accounts of the two departments, the factory and the selling department, should be kept separate.

The Author. Mr. Jackson's comment on Par. 12 shows that he is in accord with me as to the distinction between production and selling. It seems immaterial whether we call these divisions processes or functions.

He contends that it is not often the case that the selling price of a manufactured product remains more uniform than the prices of the materials entering into its manufacture. Possibly this is true when the product is simple and staple; it is not true, however, as to specialized products, especially those controlled by patents or trademarks, the prices of which frequently remain unchanged for months or years, whereas the prices of the materials from which they are made usually fluctuate weekly or daily.

I do not regard "sales" as a proper basis for the distribution of commercial or selling expenses. In the case of staple products prices fluctuate frequently, sometimes daily, and in the case of nearly all products different prices are charged for the same article at the same time to different classes of customers. The price basis, therefore, is a fluctuating one, whereas the cost basis is substantially constant, the changes in it taking place gradually and infrequently. I believe that on full consideration of all the factors it will be found that "shop cost" constitutes the most logical, most stable, and most accurate basis for the distribution of commercial expenses.

Mr. Smith suggests that the importance of the subject justifies the appointment by the Society of a special committee to formulate and submit a standard system for computing manufacturing costs. I heartily concur in this suggestion. Already, in several lines of industry with which I am familiar, there is a movement among the manufacturers concerned to bring about uniformity in methods of cost accounting, in order thereby to promote closer uniformity in quotations. Experience has shown that divergence in quotations is frequently based upon divergence in

costs, due in turn to divergence in methods of computing costs. The Society can render no greater service to organized industry than by taking the lead in bringing about a reform in the method of computing costs, and in establishing a standard method for the accomplishment of this vitally important part of the work of industrial management.

Mr. Smith questions the correctness of my views concerning the distribution of interest on capital. He says "interest on borrowed capital has to be paid whether or not there is any profit in the business." This may be true also as to dividends. A dividend is sometimes paid (out of accumulated profits) when it has not been earned, and still more frequently the dividend actually paid represents only a fraction of the profit earned. In both cases the question relates merely to the disposition of profits, and to the different ownership of capital involved. Mr. Smith would amend Axiom 15 to read "Interest on borrowed capital should be treated as an operating expense." To this I must dissent. It is immaterial who owns the capital employed in a business, and whether it is represented by stock, by bond, or by bills payable. All of it should be treated alike for accounting purposes. Either no deduction should be made from profits on account of interest on capital, or else interest should be deducted on the entire amount of capital employed. Any other division would imply differences between the results obtained in one case and in others which would be fatal to the establishment of a standard of comparison.

I endorse unqualifiedly Mr. Smith's closing statement that the main thing is to work for a standard system.

No. 1378

THE PRESENT STATE OF THE ART OF INDUSTRIAL MANAGEMENT

MAJORITY REPORT OF SUB-COMMITTEE ON ADMINISTRATION

During the past few years a number of striking phenomena, in connection with industrial management, must have become evident even to the most superficial observer. The more important are:

a The widespread, popular interest in the subject which had its rise in a statement made before the Interstate Commerce Commission, in a hearing on the matter of proposed advances in freight rates by carriers. An attorney for the shippers stated on November 21, 1910, that it was estimated that by the application of newly discovered principles of management "in the railroad operation of this country an economy of $1,000,000 a day is possible," and further that these principles can be applied with equal success "in every form of business activity." This popular interest is shown by the great number of articles published in the daily papers and popular magazines, mediums that give but scant attention to technical subjects, except of the most striking nature.

b The suddenly intensified interest in the subject on the part of employers and business executives in many lines of activity, shown by lectures, addresses, professional papers and reports presented to their associations.

c The opposition of labor unions to the newer methods of management, shown by statements of labor leaders, in a few instances by strikes, and by an attempt to

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prohibit by law the use of some of these methods in Government shops.

d Governmental recognition of the matter shown by the appointment of a special committee of the House of Representatives to investigate systems of management in Government arsenals and shops, which reported in March 1912; by the appointment of a civilian board by the Secretary of the Navy to investigate management in the navy yards, which reported in July 1911; and by Senate bill *S 6172*, now in committee, which is intended to prohibit time study and the payment of premiums or bonus on Government work.

e The rapidity with which literature on the subject has accumulated. One directory of books on business management lists 500 titles, and states that 75 per cent of them have been written within five years.

f The formation of two societies having as an aim the furtherance of the application of the principles of management.

g The separation of persons interested in the matter into two camps, one of enthusiastic advocates, the other of vigorous opponents of what is called the new element in management.

h The unquestionable proof of the advance that can be made in unskilled work, as shoveling material, and in ancient trades, as bricklaying, by the application of the principles of management. This is the most striking phenomenon of all.

THE PRINCIPLES OF MANUFACTURE

2 Before defining the element in the art of management that has given rise to these phenomena, it is necessary to review briefly the beginnings of modern industry. This gives a historical setting from which the present can be more truly judged.

3 Modern industry is stated by some writers to have begun in 1738 when John Wyatt brought out a spinning machine. Others place the period as between 1750 and 1800, when the power loom and steam engine came into being. It was marked by the development of labor-saving machinery. It was brought about by the change from handicraft to manufacture.

4 Early British economists held that the application of the principle of *division of labor* was the basis of manufacture. From Adam Smith's "Wealth of Nations," 1776, we quote:

"This great increase of the quantity of work which, in consequence of the division of labor, the same number of people are capable of performing, is owing to three different circumstances; first, to the increase of dexterity in every particular workman; secondly, to the saving of the time which is commonly lost in passing from one species of work to another; and lastly, to the invention of a great number of machines which facilitate and abridge labor, and enable one man to do the work of many."

5 Charles Babbage, the great British mathematician and mechanician, believed that from the above-quoted statement the most important principle was omitted. This omission he supplied as follows in his "Economy of Machinery and Manufacture," 1832:

"That the master manufacturer, by dividing the work to be executed into different processes, each requiring different degrees of skill and force, can purchase exactly that precise quantity of both which is necessary for each process; whereas, if the whole work were executed by one workman, that person must possess sufficient skill to perform the most difficult, and sufficient strength to execute the most laborious, of the operations into which the art is divided."

6 It appears, however, that another principle is the basic one in the rise of industry. It is the *transference of skill.* The transference of skill from the inventor or designer to the power-driven mechanism brought about the industrial revolution from handicraft to manufacture. It will be necessary to refer to this principle frequently throughout this report, in showing the meaning and position of management in industry.

7 No better single illustration of the application of this principle can be found than in the invention of the lathe slide rest by Henry Maudsley in 1794. This has been ranked as second only to the steam engine in its influence on machinery building, and thus on industrial development. The simple, easily controlled mechanical movements of the slide rest were substituted for the skilful human control of hand tools. So complete has been this transference of skill that today hand tooling is a vanished art in American machine shops. Very few lathe hands can chase a thread with hand tools, yet all can cut good threads on an engine lathe, thanks to the slide rest. After the traditional skill of a trade, or the special, pe-

culiar skill of a designer or inventor, has been transferred to a machine, an operator with little or no previously acquired skill can learn to handle it and turn off the product.

8 An example of the extent to which this transference of skill is carried today is presented by the shoemaking industry. The United Shoe Machinery Company builds some 400 machines used in shoe manufacture. These are so highly organized that the greater part of shoe-shop operatives are unskilled except in a single readily mastered detail of the work. The skill in shoemaking is now in the mechanical equipment of the shops. This transference is a development of the past 50 years.

9 James Nasmyth, a British engineer, inventor of the steam hammer, has this to say in 1851 of the application of this principle in his own works: "The characteristic feature of our modern mechanical improvements, is the introduction of self-acting tool machinery. What every mechanical workman has now to do, and what every boy can do, is not to work himself, but to superintend the beautiful labor of the machine. The whole class of workmen that depend exclusively on their skill is now done away with."

10 Methods of analyzing and recording operations were early developed. Adam Smith records the divisions of the work of manufacturing pins, listing 11 operations. Charles Babbage gives a table (see Appendix No. 1, Table 1) from a French investigator, showing the number of operations, time for each, cost of each, and expense of tools and material for making pins in France in 1760. He gives a similar table for English manufacture in his day (see Appendix No. 1, Table 2, from Economy of Machinery and Manufacture, 1832).

11 He further comments on the use of the watch to time operations. We quote from his instructions to one making such observations and using a skeleton form that he recommends: "In filling up the answers which require numbers, some care should be taken: for instance, if the observer stands with his watch in his hand before a person heading a pin, the workman will almost certainly increase his speed, and the estimate will be too large.....The number of operations performed in a given time may be frequently ascertained when the workman is quite unconscious that any person is observing him. Thus the sound made by the motion of a loom may enable the observer

to count the number of strokes per minute, even though he is outside the building in which it is contained."

12 M. Coulomb, the noted French physicist (1736-1806), who had great experience in making such observations, cautions those who may repeat his experiments against being deceived by such circumstances. We translate a single quotation: "I pray (says he) those who wish to repeat them (the experiments) if they have not time to measure the results after several days of work, to observe the workmen at various times during the day without their knowing that they are being watched. We cannot be too well warned of the danger of self-deception in computing either the speed or the effective time of work through an observation of a few minutes."

13 Thus we see the application of the principle of *transference of skill* at the basis of the development of the industry, and an early appreciation of the value of the detailed study of operations in making that transference more complete. But the machine was the viewpoint. It was looked upon as the producing unit. Combined and contrasted with this was a lack of knowledge of scientific principles and their sure application. Charles Babbage treats of this forcefully. We quote:

> "There is perhaps no trade or profession existing in which there is so much quackery, so much ignorance of the scientific principles, and of the history of their own art, with respect to its resources and extent, as is to be met with amongst mechanical projectors."

14 In the same vein he emphasizes the need of accurate drawings as if having in mind the poor quality of the work from the average draftsman of his day: "It can never be too strongly impressed upon the minds of those who are devising new machines (says he) that to make the most perfect drawings of every part tends essentially both to success of the trial, and to economy in arriving at the result."

15 He further points out that there is another important factor in successful industry, in addition to machinery. We read that "in order to succeed in a manufacture, it is necessary not merely to possess good machinery, but that the domestic economy of the factory should be most carefully regulated."

16 These quotations foreshadow modern methods of thinking out the work in advance and transferring this thought to the workmen. The subsequent development has had the effect

of advancing still further the division of labor, and beginning the division of thought. The drafting room presents the first example of the trend, in its collection of engineering data, in its prediction of results and the formation of staff organization.

17 But from the period of the last quotation almost to the present there has been no change in the basic principles discovered and applied in industry. There has been nothing but an extension of those already known. The place of greatest advance has been in the drawing room. The art of machine design has been greatly developed. The last half of the last century saw a tremendous increase in inventions, a tremendous furtherance of the application of transference of skill to machines and tools. The skeleton of an industrial organization of this period, one that was too large for a single executive to manage, consisted of a designing department and a production department, each with a head responsible to the manager.

18 The first of these, the one that was the means of embodying skill in the machinery and tools of production, was highly developed and organized. Experiment, research and detailed study were constantly resorted to, to aid in reaching the desired result. The work was highly specialized and the employees highly paid. Not infrequently the manager or chief executive devoted much of his own time to this part of the business.

19 The production department presented a contrasting condition. The workmen were given the tools and machines designed in the drawing room and using their own unaided skill were expected to produce work of the desired quality and quantity. Except in rare instances no effort was made to transfer the skill of the management to the production department and the employees, or to undertake the division of executive thought. Very little consideration was given to the workmen as a producing unit.

FEATURES OF THE CHANGE

20 Within the past 20 or 25 years certain changes have taken place in the attitude of many production managers toward the problems that they face and the forces and means that they control. An increasing amount of attention is being given to the worker. An early evidence was the development of profit-sharing, premium and bonus systems to reward increased effort

and output. There followed welfare work, industrial betterment movements, the adoption of safeguards and regulations to minimize industrial accidents, the substitution of the principle of accident compensation for employers' liability and an improvement in the physical surroundings and conditions of factories. All of these tendencies have been fostered and to a great extent initiated by employers. But even today these are by no means generally adopted.

21 Another tendency, less pronounced in character, has as its object the improvement of the personal relations between employé and employé and between employé and employer. It is an effort to establish the best of factory working conditions in those things not physical in nature, to develop and maintain a shop atmosphere free from all harassing and hindering influences. It is an attempt to make use of the results of experimental psychology, in improving working conditions.

22 But the most important change and one that comprehends the others, is in the mental attitude toward the problems of production. The tendency is toward an attitude of questioning, of research, of careful investigation of everything affecting the problems in hand, of seeking for exact knowledge and then shaping action on the discovered facts. It has developed the use of time study and motion study as instruments for investigation, the planning department as an agency to put into practice the conclusions drawn from the results of research, and methods of wage payment which stimulate coöperation.

23 All of these changes have affected the production department much more than the designing department. The effect is to extend the principle of transference of skill to production, so that it completely embraces every activity in manufacture. The skill of the management is consciously transferred to all of the operations of the factory. This extension is expressed by these phrases: the drawing room is the planning department of design, and the planning department is the drawing room of production.

NATURE OF THE COMMITTEE'S INVESTIGATION

24 To obtain information on present conditions your committee wrote to the recognized experts, to executives of plants in many lines of industry, to students of industrial problems,

and has had many interviews with men in these various fields. The response to our requests has been in the main most generous. We are deeply indebted to the information thus received for a large portion of the following sections of this report. We are glad to take advantage of this opportunity to express our gratitude to all those who have given aid.

25 Throughout the following pages there is a plentiful use of illustrative quotations. Many of these are taken from correspondence resulting from our investigations. Others are from the mass of literature mentioned in Paragraph 1*e*.

26 On some points diametrically opposed views have been expressed. In such cases we have presented both. In no case has credit been given for these views or quotations, as the information was solicited in confidence.

DEFINITION OF THE NEW ELEMENT IN THE ART OF MANAGEMENT

27 Requests for a definition of the new element in the art of management brought forth a difference of opinion as to its existence. The opposed view is given in the following quotations:

"I am not aware that a *new* element in the art of management has been discovered———."

"There have been no new discoveries in scientific management of industrial institutions. Common-sense men have used common-sense methods always. The term 'scientific management' is a catch-word which assumes that industrial institutions have not been scientifically managed—which is not the case. My experience and the experience of my friends has been that there has been no new element injected into the art of management."

"In the writer's opinion there is very little that is new about it (the art of management). There is hardly any part of it that has not been practised by managers for the past 100 years. The trouble is there are not enough managers with sufficient initiative to set the system moving properly."

"——— the problem presented is not the adoption of something entirely new; but rather the extension to every detail of our work of something which we have already tried."

28 Turning now to the other side of the question, from a large number of definitions of this new element we select the following as very nearly conveying, taken together, the complete conception as our investigation has disclosed it:

"The best designation of the new element I believe to be 'scientific management.' This term already has been adopted quite generally and although frequently misused, carries with it the fundamental idea that the manage-

ment of labor is a process requiring thorough analytical treatment and involving scientific as opposed to 'rule of thumb' methods."

"The writer ventures to define the new element briefly, but broadly, as: The critical observation, accurate description, analysis and classification of all industrial and business phenomena of a recurring nature, including all forms of coöperative human effort and the systematic application of the resulting records to secure the most economical and efficient production and regulation of future phenomena."

"Stripped of technicalities the method of the modern efficiency engineer is simply this: First, to analyze and study each piece of work before it is performed; second, to decide how it can be done with a minimum of wasted motion and energy; third, to instruct the workman so that he may do the work in the manner selected as most efficient."

"The Taylor System is not a method of pay, a specific ruling of account books, not the use of high-speed steel. It is simply an honest, intelligent effort to arrive at the absolute control in every department, to let tabulated and unimpeachable fact take the place of individual opinion; to develop 'team play' to its highest possibility."

"As we conceive it, scientific management consists in the conscious application of the laws inherent in the practise of successful managers and the laws of science in general. It has been called management engineering, which seems more fully to cover its general scope than a science."

29 These quotations convey the ideas of a conscious effort to ascertain and study facts and systematically to apply them in instructing the workmen and in controlling every department of industry. Setting these against the underlying principle of the transference of skill we conceive the prominent element in present-day industrial management to be: *The mental attitude that consciously applies the transference of skill to all the activities of industry.*

30 Here emphasis is placed upon the word *all* for, as shown in Pars. 17 and 18, the restricted application of this principle to machines and tools has been highly developed for a long period. But its conscious application in a broad way to the production departments, and particularly to the workmen, we believe has been made during the last quarter century.

RISE OF THIS MENTAL ATTITUDE

31 The rise of this change of attitude in regard to industrial management is shown in the papers on the subject in the Transactions of this Society. These are 16 in number and are listed in Appendix No. 2 of this report. The period covered is from 1886 to 1908. The practice upon which several were based

extended over a number of years before the paper was presented. Papers on accounting have been excluded.

32 The first, No. 207, classifies *management of works* as a modern art having a vast amount of accumulated experience, points out that the executives must have "a practical knowledge of how to observe, record, analyze and compare essential facts in relation to ..... all..... that enters into or affects the economy of production and the cost of the product," and makes a plea for the interchange of management data.

33 Eight following papers, Nos. 256, 341, 449, 596, 647, 928, 965 and 1012, deal with methods of wage payment, showing the increasing attention given to the workmen during this period. Of these methods the "premium plan" described in paper No. 449 has an extensive use today in machine shops. It probably ranks third, in extent of use, being exceeded by day work and piece work in the order named. Paper No. 647 outlines elementary rate fixing; that is, the minute study of each detail of each operation. From this, motion study and time study have grown. The "bonus system" of paper No. 928 also has an extensive use, probably ranking fourth.

34 Paper No. 1003, "Shop Management," is the first complete presentation of the subject. This paper with the subsequent writings of its author, stands today as the only comprehensive outline of industrial management. Papers Nos. 1001, 1002, 1010, 1011 and 1115, are amplifications of certain features of No. 1003 and are based on the same practice.

35 Paper No. 1221 deals with the training of workmen, and outlines practical, tested methods of bringing about the all-important transference of skill.

LABOR-SAVING MANAGEMENT

36 Since these papers were presented, and during the development of popular interest in the subject, the term "scientific management" has been generally and loosely applied to the new system and methods. This is commonly taken to mean that there is a science rather than an art of management. A truer interpretation is that it means management using scientific methods, these being taken largely from the sciences of physics and psychology.

37 The expression "labor-saving management" better con-

veys the meaning of the movement. It has the further advantage of being easily and surely understood because of its strict analogy with the term "labor-saving machinery." It is no chance that puts these two terms labor-saving machinery and labor-saving management, in conjunction, for the first is the past development and the second the present trend of industry, and they will be closely and inevitably associated in the successful manufacturing of the future. Throughout the following pages of this report the terms "industrial management" and "labor-saving management" are used, the first to denote the subject broadly, the second the newer attitude.

THE REGULATIVE PRINCIPLES OF INDUSTRIAL MANAGEMENT

38 The lack of accurate thinking and clear expression in regard to management are nowhere better shown than in many of the statements of the so-called principles. These can be divided into two classes, personal characteristics of managers and mechanical means of applying. It is evident that neither can show us the way in which the activities of industry are to be regulated.

39 In our investigation preparing for this report, one correspondent writes as follows:

"The regulative principles of management along scientific lines include four important elements:

a Planning of the processes and operations in detail by a special department organized for this purpose.

b Functional organization by which each man superintending the workman is responsible for a single line of effort. This is distinctly opposed to the older type of military organization, where every man in the management is given a combination of executive, legislative and judicial functions.

c Training the worker so as to require him to do each job in what has been found to be the best method of operation.

d Equable payment of the workers based on quantity and quality of output of each individual. This involves scientific analysis of each operation to determine the proper time that should be required for its accomplishment and also high payment for the worker who obtains the object sought."

40 Another correspondent finds the solution of problems of management in the observing and regulating of three classes of industrial phenomena:

"*a* The economic results of different arrangements and forms of

materials and operations upon them, either to produce equipment or product. This covers the whole field of recorded experience from invention and design of product and tools down through the successive shop processes to ultimate finished product and its tests in service. It is the object of the scientific method to make the best of this experience, in its essential details, readily available for all concerned, and to see that it is actually absorbed and put in practice.

b The economic results of varying executive methods for effectively directing human efforts as a whole in the use of the above experience. This covers the entire field of building up, coördinating and controlling the supervising organization of a plant with its statistical and recording systems.

c The economic results of steps taken to raise the industrial efficiency of the individual worker in every grade of service. This covers the whole problem of labor reward, intensified ability, conserved energy and the general relations of employer and employée."

41 We have pointed out that the underlying principle, that is, cause in the widest sense, the application of which has built up modern industry, is the transference of skill. This basic principle is put into effect on the management side of all industrial activities, through three regulative principles which sum up the ideas in the above quotations, Pars. 39 and 40. These have been concisely stated as[1]: (*a*) the systematic use of experience; (*b*) the economic control of effort; (*c*) the promotion of personal effectiveness.

42 The first includes the use, in all essential detail, of traditional knowledge, personal experience and the results of scientific study on the part of the executive force. It implies the accumulation and use of records and the setting up of standards.

43 The second includes the division and subsequent coördination of both executive and productive labor; the planning of single lines of effort, the setting of definite tasks and the comparison of results; and the effective training of the workers. It implies the previous acquisition of skill by the executives.

44 The third includes a definite allotment of responsibility and the adequate, stimulative encouragement and reward of both executive and productive labor; the development of contented workers, and the promotion of their physical and mental health. It implies the most thorough comprehension of the human being.

[1] American Machinist, vol. 36, p. 857, The Principles of Management, by Church and Alford.

THE PRACTICE OF MANAGEMENT

45 As labor-saving management springs from a change in mental attitude, the beginning of its practice should be with the persons having the final responsibility, the proprietors of closely-owned businesses, the directors of larger establishments, or the officials having charge of Government works. Before any changes are made, such men should clearly understand the viewpoint from which all of the managerial work is to be done, the principles that are to be applied, the general method of their application and the results expected.

46 A similar mental attitude must be fostered among all the members of the executive force and a period of training for them begun. This may include a redistribution of function and responsibility, and will include a detailed study of production by scientific methods. This is the period of division of thought, training of the management staff and setting-up standards of performance. This must be carefully performed before there can be effective transference of skill to the workers in the production departments.

47 The usual conception of modern management is that it affects the workmen most of all, tending to stimulate them to turn out increased production to their possible hurt. This is wrong. If the principles outlined are followed, the executive, or non-producing labor is the most affected. Its individuals are compelled to study, plan and direct. They must acquire knowledge and skill in order to transfer it. It is a system of management that forces the executives to manage.

48 This being so, the introduction of modern management in a plant must be made slowly. The causes of most so-called failures are principally two: a failure of the executives to acquire the vital mental attitude and too great haste in application. The latter seems to be the dominant one. Your committee feels compelled to emphasize the danger of attempting to hurry any change in methods of management. Each step of the work should be made permanent before the next is begun.

49 We have examined records of production which clearly show a lessening of individual output among workers who had been trained for some time and had achieved good results as soon as untrained workers were put with them, thus lessening their share of personal supervision. Later the original standard

of production was again reached, but the results seemed to be directly proportional to the amount of skilful supervision, during a lengthy period of training.

50 After those who are to operate the new methods have acquired the necessary knowledge and established sufficient standards, the work of putting these into effect can be begun. This means the fixing of the best attainable working conditions and giving each worker definite tasks with an adequate reward to each one who attains to the standard set. This part of installing the methods must be accomplished with tact and patience, remembering that leadership and example are powerful aids in bringing about enthusiastic coöperation.

51 The training of the workers is essential in this part of the application. This must be far more than mere demonstration, the mere showing that a thing can be done. It must be patient teaching and help until the required degree of dexterity or skill is acquired, that is, up to the habit stage. It is evident that such work cannot be hurried.

52 Such, broadly, are the three steps in the practice of management. It is now necessary to investigate the internal elements of permanence in such methods. If the proper mental attitude is once taken, we believe it will never be given up. This is substantiated by a few cases when early attempts to improve management were failures and the methods abandoned. Later, however, other attempts were made with substantial success. The mental attitude outlived the failure. Thus in a given industrial organization this feature would not be lost except by a loss of the executive staff.

53 The permanence of records of performance and standards needs only to be mentioned to be appreciated. Once set up in an industry, disaster is invited if they are disregarded.

54 To these is added a third in the nature of a spur from the working forces to the managing force. An adequate reward is one of the essentials. Whatever disturbs the mechanism of production interferes with the earning of the rewards. The workers at once object, pointing out the trouble and insisting that it be rectified. The management is spurred to keep all conditions up to the fixed standard. Examples of this action have been brought to the attention of your committee.

55 The practice as outlined, while built upon fixed standards

and procedure, is by no means rigid and inflexible as has been alleged. The design and construction of labor-saving machinery is carried on with a multiplicity of different details. Labor-saving management should likewise use a variety of details suited to the requirements of different industries and plants. There can be nothing fixed in such human endeavor except the underlying principle. As a simple matter of fact we have found different methods, details and nomenclature in use in different plants. Many efforts have undergone marked change and development since first installed. Further, this idea of rigidity is repudiated by some of the foremost management experts.

56 In Par. 39 is emphasized the need of a scientific study of everything connected with production. The methods used are adapted from the research laboratory. But the purpose of their use is changed. The scientific investigator uses his laboratory to discover facts. Their discovery and declaration is his end and aim. The management investigator uses laboratory methods to discover facts for immediate use. The end and aim is utility. This is the test of industry. It is therefore unwise and in fact detrimental to carry investigations to an extreme. Enough facts must be observed to shape intelligent action. Persons having time study and motion study in charge should possess that rare, intuitive, human quality that causes its possessor to know when enough observations have been collected to form a sound working conclusion.

57 The position of the expert in the practice of management is more clearly seen as experience increases. The element of mystery has already departed. This is to be welcomed for it means the downfall of mere "systematizers." One of the unfortunate features of this great movement has been the rise of alleged experts who have been ready to promise extravagant results if they were allowed to systematize an industrial plant. The test which their work cannot meet is the one of permanence.

58 An industrial manager who has had signal success in directing large enterprises sums up the more undesirable characteristics of systematizing practice as:

"*a* The publication and quotation of statistics regarding gains made through the use of particular systems, without a frank statement of the degree of inefficiency of the plants before reorganization.

b The failure to view the plant from the investor's standpoint rather than as a laboratory offering opportunities for interesting and expensive experience.

c The failure to admit that every application of past solutions to unstudied new and different conditions *is* an experiment.

d The waste of time and money on problems that will yield to scientific treatment, but which do not recur often enough to justify such a solution.

e The undervaluing of effective leadership in management and consequent lack of permanency in results.

f The overvalue of emasculated 'system' leading to a curious nonresponsibility on the part of any person for the total result.

g The frequent assumption that the treatment of the problems of similar plants should be identical.

h The failure to properly appraise in a growing concern the value of its internal asset of 'good will.'

i The imperfect analysis and appreciation of the human factor in industry, with a consequent failure to reckon patiently with 'habit' and 'inertia' and a tendency to hasty 'substitution,' bringing about the breaking up of valuable organization."

59 The real expert concentrates on the facts of a given problem, and from a wide experience in analysis, coördination and practical responsibility works out a solution by scientific methods, suited to the material and human factors involved. The tendency is for him to do less of the detail work of installation, but to train and direct the persons who are permanently to manage. This is a true process of transference of skill.

STATISTICAL DATA

60 Your committee hoped to present statistics on the extent to which labor-saving management is in use. This could not be realized. Many industrial managers whom we have addressed have not honored us with their confidence in this direction. In fact, it seems as if a secretive stage is now with us. There are two reasons for withholding such information. The first is identical with the one that has developed "trade secrets" and secretiveness in regard to machines, tools and processes, the desire to keep things of value away from competitors. The second is a belief that in the minds of some persons a reflection is cast upon the ability of the executives of an industrial establishment if outside experts are employed. Frequently a system of management is referred to as the development of some one in the organization, although it was installed by a management expert, employed for the purpose.

61 Some idea of the variety of the industries in which labor-saving management is in use can be gained from Appendix No. 3, which lists a total of 52.

BROAD RESULTS OF LABOR-SAVING MANAGEMENT

62 In cases where the use of labor-saving management can be considered a success, the broad results have been: a reduced cost of product; greater promptness in delivery with the ability to set and meet dates of shipment; a greater output per worker per day with increased wages; and an improvement in the contentment of the workers. This last item is shown by the fewness of strikes under the new management, and in the refusal of those working under the changed conditions to join in a strike of their fellows in the same plant who were not working under the new methods. This last-mentioned situation has arisen a number of times. In one case an attempt was made to strike a room where about one-half of the operators were under the new conditions. These refused to go out; the rest went.

63 These results indicate certain advantages to both employer and employé. But it is charged that the movement has not yet entirely justified itself from the economic viewpoint, for it has not reduced the cost of product to the consumer. The implication is that its possibilities will not be realized until employers, employés and the public are alike benefited. With this view we are in most hearty accord. Labor-saving machinery has brought the comforts that we all enjoy today. Labor-saving management promises to extend those comforts. Where properly administered it is conserving labor and is thus contributing to the good of society at large, and although the benefit to the consumer may not yet be generally felt, it has already developed to a certain extent and will continue to develop as the natural result of increased production.

J. M. Dodge, *Chairman*
L. P. Alford, *Secretary*
D. M. Bates
H. A. Evans
Wilfred Lewis
W. L. Lyall
W. B. Tardy
H. R. Towne

Members Sub-Committee on Administration

APPENDICES

APPENDIX NO. 1

64 The following tables are taken from a book published in 1832, Economy of Machinery and Manufacture, by Charles Babbage. Table 1 gives the cost and operations in detail in manufacturing 12,000 No. 6 pins in France in 1760. The observations were made by M. Perronet.

65 Table 2 gives similar data for manufacturing pins in England in the time of the author. The size is "eleven" of which there are 5546 to the lb.

TABLE 1 OPERATIONS IN PIN MANUFACTURE IN FRANCE IN 1760

	Name of the Process	Time of Making 12,000 Pins, Hr.	Cost of Making 12,000 Pins, Pence	Workman usually Earns per Day, Pence	Expense of Tools and Materials, Pence
1	Wire	....	....	....	24.75
2	Straightening and Cutting	1.2	0.5	4.5	....
3	Coarse Pointing	1.2	0.625	10.0	....
	Turning Wheel *	1.2	0.875	7 0	....
	Fine Pointing	0.8	0.5	9.375	....
	Turning Wheel	1 2	0.5	4.75	....
	Cutting off Pointed Ends.	0.6	0.375	7.5	....
4	Turning Spiral	0.5	0.125	3.0	....
	Cutting off Heads	0.8	0.375	5.625	....
	Fuel to Anneal Heads	....	....	....	0.125
5	Heading	12.0	0.333	4.25	....
6	Tartar for Cleaning	....	....	....	0.5
	Tartar for Whitening	....	....	....	0.5
7	Papering	4.8	0 5	2.0	....
	Paper	....	....	....	1.0
	Wear of Tools	....	....	....	2 0
		24.3	4.708		

* The expense of turning the wheel appears to have arisen from the person so occupied being unemployed during half his time, whilst the pointer went to another manufactory.

TABLE 2 OPERATIONS IN PIN MANUFACTURE IN ENGLAND ABOUT 1830

	Name of the Process	Workmen *	Time of Making 1 Lb. of Pins, Hr.	Cost of Making 1 Lb. of Pins, Pence	Workman Earns per Day	Price of Making Each Part of a Single Pin in Millionths of a Penny
					s. d.	
1	Drawing Wire	Man	0.3636	1.2500	3 3	225
2	Straightening the Wire	Girl	0.3000	0.1420	0 6	26
		Woman	0.3000	0.2840	1 0	51
3	Pointing	Man	0.3000	1.7750	5 3	319
4	Twisting and Cutting the Heads	Boy	0.0400	0.0147	0 4½	3
		Man	0.0400	0.2103	5 4½	38
5	Heading	Woman	4.0000	5.0000	1 3	901
6	Tinning, or Whitening	Man	0.1071	0.6666	6 0	121
		Woman	0.1071	0.3333	3 0	60
7	Papering	Woman	2.1314	3.1973	1 6	576
			7.6892	12.8732		2320

* Number of Persons Employed: Men, 4; Women, 4; Children, 2. Total, 10.

APPENDIX NO. 2

66 Following is a complete list of papers published in the Transactions of The American Society of Mechanical Engineers dealing with industrial management:

No.			
207	The Engineer as an Economist	Henry R. Towne	1886
256	A Problem in Profit Sharing	Wm. Kent	1887
341	Gain Sharing	Henry R. Towne	1889
449	The Premium Plan of Paying for Labor	F. A. Halsey	1891
596	The Relation of the Drawing Office to the Shop in Manufacturing	A. W. Robinson	1894
647	A Piece-Rate System	Fred. W. Taylor	1895
928	A Bonus System for Rewarding Labor	H. L. Gantt	1902
965	Gift Propositions for Paying Workmen	Frank Richards	1903
1001	The Machine-Shop Problem	Charles Day	1903
1002	A Graphical Daily Balance in Manufacture	H. L. Gantt	1903
1003	Shop Management	Fred. W. Taylor	1903
1010	Slide Rules for the Machine Shop as a Part of the Taylor System of Management	Carl G. Barth	1904
1011	Modifying Systems of Management	H. L. Gantt	1904
1012	Is Anything the Matter with Piece Work?	Frank Richards	1904
1115	A History of the Introduction of a System of Shop Management	James M. Dodge	1906
1221	Training Workmen in Habits of Industry and Coöperation	H. L. Gantt	1908

APPENDIX NO. 3

67 Following is a list of the industries in which some form of labor-saving management has been installed:

Book binding
Building construction
Carriage and wagon building
Construction and repair of vessels (navy yards)
Fire-arms and ordnance
  Rifles
  Gun carriages
Machinery building
  Automobiles
  Agricultural implements
  Coal-handling machinery
  Electrical machinery
  Founding, iron and brass
  General machine work
  Gas engines
  Locomotives
  Machine tools
  Molding machines
  Pumps
  Pneumatic toools
  Sewing machines
  Typewriters
  Wood-working machinery
Metal and coal mining
Metal working
  Bolts and nuts
  Chains
  Hardware
  Tanks
  Tin cans
  Valves and pipe fittings
Miscellaneous manufacturing
  Beer
  Beet sugar
  Boxes (wood and paper)
  Buttons
  Clothing
  Cordage
  Food products
  Furniture
  Flour
  Glass
  Lumber products
  Pianos
  Paper and paper pulp
  Rubber goods
  Soaps
  Shoes
  Slate products
Printing and lithographing
Railroad maintenance of motive power
Steel manufacture
Textile manufacture
  Bleaching and dyeing
  Cottons
  Velvets
  Woolens

THE PRESENT STATE OF THE ART OF INDUSTRIAL MANAGEMENT

MINORITY REPORT OF SUB-COMMITTEE ON ADMINISTRATION

I am unable to sign the majority report in its entirety, much as I admire the thoroughness with which it has been prepared and its great interest. In its general tenor it distinctly implies the desirability of what is termed labor-saving management, involving the planning department, functional organization and the bonus system. Perhaps this statement is not strictly justified, but I cannot avoid the impression after reading the report most carefully. That the methods of management have undergone a great change in recent years I certainly agree. I would explain it by stating that in many respects the art of management is developing into a science. In common with most lines of work, the method of investigating facts has changed. Phenomena are analyzed, information is obtained accurately, the mental attitude of the manager is scientific rather than empirical. Things that used to be known generally by gradual experience are now known specifically by detailed observation. In the course of this development many new systems and ideas have been invented, time studies, motion study, payment systems, functional management, etc. Some are new, others partly new, others simply practices of many managers put into definite form. The science of management will classify these for us, perhaps even explain their advantages and limitations so that we may in time know which is preferable and when.

2 But the introduction of the use of these methods has been attended by claims as to the results that might be obtained by their use, and these claims have led to such absurd statements as that made before the Interstate Commerce Commission, which is referred to in the report. I feel most strongly that each of the suggestions that have been made to improve methods of management may have merit, but I do not feel that any one of them is a panacea for all our inefficiency. For instance, in certain classes of work time studies are valuable, in others they may be a waste of time. In certain

classes of work I consider piece-work or bonus systems desirable, in others I consider them inferior to day work. Functional management may be an improvement in certain industries, in others I do not consider it suitable. In some cases the workman can be trained by the skilled executive, in others he can train him with ease.

3 In general I feel that labor-saving management is not any particular system, but will always remain the art of selecting and applying the most appropriate methods furnished by the science of management, the science that records what these methods are and the results obtained from them. As you will see, I agree with your correspondent in his last paragraph, quoted under Par. 28; but his opinion is not really represented in the trend of the report.

H. H. Vaughan
Member, Sub-Committee on Administration

DISCUSSION

CHARLES B. GOING expressed the view that there was a lack of correspondence between the title and the substance of the majority report. His impression from reading the report was that, after an introductory attack upon the analysis of the elements of industrial management, it proceeded to argumentation on its own behalf and presented, not a survey of the present state of the art of industrial management as it exists in this country, but an interpretation of a theory of management as it exists in the convictions of the majority of the committee. He did not intend to depreciate the interest of the conclusions expressed by the majority of the committee, but only to point out that these do not constitute a report on "the state of the art of management," and that they give us no more than a fragmentary idea of the conditions under which this art is carried on in the United States today.

A most commendable stand is taken by the committee in using the term "scientific management" descriptively, and not in the titular sense, applicable only to the Taylor system, in which it has often been employed. Scientific management should mean, as the committee says (Par. 36), "management using scientific methods, these being taken largely from the sciences of physics and psychology." But even with this definition in mind, the committee appear to have limited their views too closely to questions of physics and to have given inadequate attention to the questions of psychology, which, in the estimate of many students of the subject, are really paramount in the art and practice of industrial management.

Scientific management, using the term now in its titular sense, offers one method of finding standard individuals. There are other methods, forming no part of the systematic ritual of this school, which are stated to be far more effectual in discovering and distributing standard individuals. There are policies or methods of much interest, designed to better the conditions of operation with the apparently large numbers of non-standard or sub-standard individuals by whom so much of the work of industry is now carried on. Quantitatively, these schools of thought and of practice are of much importance in the existing state of the art of industrial management. The Taylor system is but one factor. There is much valuable contribution to thought and practice outside of the list of publications published in the Society's Transactions and listed in Par. 66.

By leaving this apparently out of its consideration, and concentrating its attention upon a single factor instead of analyzing the entire equation, he thought the majority report of the committee was but another contribution to an already overburdened controversy, and that it had left the general survey not only unfinished, but scarcely even begun.

Halbert P. Gillette, while able to subscribe to many of the statements of the report, took exception to the statement that "This paper (Shop Management by Fred. W. Taylor) with the subsequent writings of the author stands today as the only comprehensive outline of industrial management."

Mr. Taylor's admirable paper on Shop Management and his subsequent writings (which are all restatements of the principles given in that paper) are so far from being a "comprehensive outline of industrial management" that one needs but to contrast the four "Taylor principles" of management with the twenty or more principles that have already been formulated to recognize that Mr. Taylor has confined his writings to a very narrow, though very important class of industrial management principles.

In 1908, in conjunction with Richard T. Dana, the writer prepared a series of lesson papers entitled Cost Analysis Engineering which were published in book form and in which appears what the writer believes to be the first use of the term "science of management." In that book six principles of management were formulated.

In 1909, Cost Keeping and Management in Engineering, by Gillette and Dana, were published and dedicated to Fred. W. Taylor. This book contains an execution of part of the writer's original idea of writing a philosophy of management. Ten laws of management are formulated in this volume and seventeen rules to be applied in managing construction work are given, several of which the authors have in subsequent writings expanded into broader generalizations which may be termed laws of management. Since writing these books the authors have prepared a considerable number of articles formulating other laws of management, such for example as the law of plant location, the best illustrations of which may be attributed to the late A. M. Wellington, whose book on The Economic Theory of Railway Location (1887) is one of the most profound exposi-

tions of the application of one law of management which has ever been written.

While to mechanical engineers belongs much of the credit for having written discussions of the principles of management, the writer believes that members of this Society have often made the serious mistake of supposing that the whole of the science of management has come within their discussions. They have quite generally overlooked the work of such giants in the civil engineering field as the late A. M. Wellington. It is true that Wellington did not write about the direct handling of men, but he so wrote about one phase of the indirect handling of men, if it might so be termed, that he is justly entitled to place as one of the greatest authorities on management engineering. Others before him have applied the principle of capitalizing operating expenses to determine which of two plants or machines is the most economic, but he applied this principle so minutely and so extensively as to revolutionize the practice of railway location. Indeed, he converted the art of railway location into a science by the application of one principle of the science of management.

The science of management embraces far more than the direct handling of men or the timing of the operations of men and machines. These may be scientific management as far as they go, but they do not comprise a science of management.

The science of management is a comprehensive code of demonstrable and formally enunciated laws for so directing the energies of men as to secure the most economic production and marketing of utilities.

The whole science of management usually covers five fields: (*a*) Directing the planning of the plant; (*b*) Directing the building of the plant; (*c*) Directing the operation of the plant; (*d*) Directing the purchase of materials and supplies for the plant; (*e*) Directing the marketing of the products of the plant.

The papers on scientific management that have been read before this Society have related, almost entirely, to the third field mentioned.

In developing a science of management, it has been the aim of Mr. Dana and the writer to apply the inductive method of logic in a thorough manner. The methods pursued by successful business men in all lines of endeavor have been collated and analyzed, the object being to draw sound generalizations from the data thus gathered. The generalizations drawn from these data have been called " laws " or " principles " of management. This

is a process that has no kinship with the invention of a new system of management. An invention, which may consist in a new combination of several old principles, is not a science and never can be properly designated as such, no matter how far reaching are the economies effected thereby.

It seems to the writer that the public in general, and even the engineering public, has thus far failed to perceive the difference between what is scientific and what is a science. Failure to recognize the fundamental difference between what is a part and what is a whole, has led to extravagant claims for the part and consequently will lead to derision of claims that may now be properly made for the whole. Such is the potency of words that many of the most serious errors arise solely because words are not strictly defined and strictly used according to definition.

A. Hamilton Church.[1] Of the two reports presented by the committee, the minority report raises the question as to whether the claims of certain groups of men to the possession of the key to all progress, have any measure of justification.

As it appears to me, the various schools of management have not presented to us any well-balanced theory of management capable of being developed in a hundred different ways according to special needs. On the contrary, it is as if they had presented to us a barrel of gunpowder, a telescope, and a magnetic compass and claimed these mechanisms to be the whole science of maritime warfare. Certain definite and concrete mechanisms such as planning departments, stop-watches, despatching boards, instruction cards of extraordinary complexity, and special methods of remunerating labor are interesting and useful, but do not make, either singly or combined, a science of management.

Moreover, the original claims, already generously broad, have been so expanded by some of the younger disciples that almost every modern method is claimed to be the direct outcome of the wisdom of one or another of the rival schools of scientific efficiency. It is because of this kind of thing that the minority report is important.

Turning to the majority report, we find that the prime underlying principle of management engineering is the transference

[1] Cons. Engr., 30 Broad St., New York.

of skill. This is an illuminative statement, and I venture to think, of great practical value. It is put very tersely in the sentence in Par. 8, "the skill in shoemaking is now in the mechanical equipment of the shop." That is a clear-cut picture of a change that is already complete, and it helps us to realize exactly what is meant by the transference of skill. It also helps us to understand why, in the engineering trades, there are such wide discrepancies in the amount of work turned out by individual operators. It is because, in these trades, the transference of skill is by no means so complete. For one thing, the capacity and range of the average machine tool is large, and to some extent indefinite. A considerable amount of skill is still vested in the worker. Still more, a remarkable amount of ignorance as to what the machine will or will not do is shared by the employer and the workman.

The report shows why this is bound to be so at the present stage. In the process of transfering hand skill to the mechanical fingers of the machine, the report emphasizes the fact that more attention has been devoted to designing it, than to the problem of using it afterwards.

The more I think over this problem, the more I am convinced that the true line of progress is the exhaustive study of machines, their capacities and limitations. I have held this opinion for many years, and the system of industrial accounting I have been advocating for the past decade was, I believe, the first step made towards bringing forward the machine to its true place as a factor of production. But I must confess that until this principle of the transference of skill was brought out so clearly by this report, I did not realize exactly why the machine tool was frequently so surprisingly ineffective under indifferent handling.

I will pass over the acceptance by the committee of the three regulative principles of management, viz: (*a*) the systematic use of experience, (*b*) the economical control of effort, and (*c*) the promotion of personal effectiveness, which were worked out by Mr. Alford and myself, except to say that the credit for the formulation of these principles belongs in a larger degree to Mr. Alford than to me, and I will conclude my remarks by calling attention to a phrase used in Par. 51 of the report, viz: "the habit stage."

All the mechanism of organization in the world is valueless

beside the steadiness of production that comes from the establishment of good habit throughout a plant. I will go further, and say that the whole object and end of organization should be to create the right kind and degree of habit in everyone of the persons engaged in production, from the president down to the shop sweeper.

It is not enough for the workman to be so instructed that he forms good habit. Every living link in the chain of production requires equally to be so trained that his acquired habit is harmonious with all the rest. The report has mentioned this aspect of the question where it insists that the executives and not the workman are the persons most important to be reached. Few people understand that the principal work of an expert organizer is not the designing of elaborate blanks and cards, but the fostering, with tireless patience, of correctly adjusted habit in each member of the staff.

As the new ideals of management engineering appear to me, and this view seems confirmed by the committee's report, they may be summed up in three sentences:

Take nothing for granted.
See that every effort is adapted to its purpose.
Cultivate habit.

These sentences are, of course, merely practical derivatives of the three regulative principles referred to in the report. Each of them in turn implies other things which will readily suggest themselves. Thus, the possibility of cultivating correctly adjusted habit depends obviously upon proper mental and physical conditions for the living forces. It implies "lead" and not "drive."

It is an interesting question where the new spirit that we find abroad in industrial management has come from. To some extent, I think, it is part of a larger movement, the realization of a sense of social solidarity, of social responsibility of each for all, that is so marked a feature of the times. But it also arises, in part, from another cause. Scientific men tell us that the great difference between a savage race and a highly civilized one is that the former remains in a condition of natural innocence, and the latter has arrived at self-consciousness. This, I think, is the real state of affairs in regard to management engineering. We are passing from a stage at which there was a simple and unconscious following of tradition, into a stage of

self-consciousness in which we are moved to subject our habits and our motives to severe self-scrutiny, and examine afresh every item of our daily practice. It is a very painful stage to have arrived at. Most of us are so content with our comfortable natural innocence that we do not like to part with it, but it is a process that once commenced, must continue.

The examination into new methods of remunerating labor, the adoption, with caution, of searching instruments of analysis, such as time study, the use of precise methods of accounting—these are not causes, but *consequences*, of this newly awakened self-consciousness. It is beginning to be recognized that production is an aggregate of infinitesimal separate acts, in each of which there are three main components. First, experience must be drawn on; secondly, the resulting effort must be intelligently adapted to the end in view; thirdly, this intelligent effort must become habitual. And to secure the successful performance of these acts, the living forces concerned must be maintained in the pink of condition, both mental and physical.

C. B. Thompson.[1] A fact not to be ignored is that labor, organized and unorganized, has taken its stand, at least temporarily, in opposition to the development of scientific management. While many of the objections raised have been so unreasonable as to be abandoned almost from the start, there are three criticisms made by working men which persist in spite of explanation: In the first place, they seem to cherish an innate resentment against time study—"putting the stop-watch on them," as they express it. It "makes them nervous," "makes them speed up unduly while under inspection," "is simply another means of slave driving," "is used unfairly," "is un-American." They say also that the method of minute planning in advance and of specific instructions as to details, is destructive of the initiative which has been the backbone of American industrial progress. Further, they assert that the enhanced product due to these methods is not fairly divided between the management and the men; that an increase of 400 per cent in production, as in the classic case of Schmidt, the pigiron handler, has been accompanied by an increase of only 60 per cent in wages.

[1] Instructor, Industrial Organization, Graduate School of Business Administration, Harvard University.

It is neither wise nor expedient to pass by these criticisms without comment. That they are all unmerited may perhaps be shown: for instance, it has been demonstrated that time study, carried on by one who is trained in the subject (and whose training is ethical as well as intellectual), is not unfair, nor is it used as a new method of driving, nor is it in any sense un-American. Time study has been carried on for decades in psychological laboratories for purely scientific purposes and with instruments even more refined than the despised stop-watch, but no one suggests that this use is open to criticism. When it is once clearly understood that time study in an industrial establishment, made under the proper conditions and by the proper persons, is aimed at similar scientific results, and that these results are then to be applied to work, subject to the mutual consent of employer and men, based on a conviction of their reliability and essential justice, these objections will disappear. But on the other hand, there is no doubt that time study can be and has been abused, and the working man has a right to be "shown."

Similarly in regard to the alleged destruction of initiative. Here the facts are clear. The machinist does not consider his initiative restrained when he is asked to do his work in accordance with the drawings supplied him by the drafting department. Systematic routing and planning of work and the development of a science by which it shall be done are, as the committee points out, analogous, for the production department, with the working drawing. We all believe in liberty, but we recognize, or are capable of being shown, that true liberty is liberty under law; that the artist is not in the least trammelled in his genius because he has been taught the laws under which he must use his materials; nor is the citizen any the less free because he, in common with all others, must conduct himself in accordance with the law of the land.

The problem of just distribution of the increased product is not capable of easy solution and demonstration, and much yet remains to be done in this field. No one has yet solved the problem of justice in distribution, and it does seem somewhat hypercritical to allege against scientific management that it has not done what any other movement or individual thus far has not been able to do. Rather it should be given credit for having pointed out the industrial necessity of justice in distribution,

and of having proposed steps in that direction. The solution of this problem is not to be looked for from engineers. When an engineer wanders into the field of economics, he is apt to make about the same diverting spectacle that the economist would who would undertake to expound the principles of machine design. Probably the wages question will be worked out in the future as it has been in the past by neither the engineer nor the economist, but by the daily struggle and adjustment known as the "higgling of the market." In this struggle the rights of the working men will have to be conserved and enforced by organization. Only by pooling their strength can they meet the superior strategic position of the employers.

The report of such a committee as this should not have overlooked the opportunity to begin or extend the campaign of education in these particulars. Something more than education is necessary, however. Labor unions are a potent and active force in present-day industry, and we should get their enthusiastic coöperation. They are the culmination of decades or centuries of development. In spite of their numerous mistakes and injustices they have, on the whole, justified their existence; and whether you agree that they have or not, they are a condition and not a theory that confronts us. It would seem to be the part of wisdom, therefore, not to take the tack of ignoring or combatting the doubts and questionings and opposition of the labor unions, but rather to persuade them of the advisability of acquainting themselves with the facts, of recognizing the inevitability of the march of labor-saving management, and to secure their active coöperation in its development. A proposal at this time to retain their positive help in the extension of scientific management may seem Utopian, but it is warranted by certain facts and precedents in the history of trade unions. They have already established their right to determine the lighting, heating and ventilating facilities of establishments in which their members work. There are cases on record in which they have required owners of old plants to scrap their obsolete machinery and install new and modern devices, in order that the owners themselves may make sufficient profit to pay adequate wages to their employees. *It is but one step, and a short one, in the extension of this principle to say to the owner that he must modernize his establishment in every detail, not because the working man is interested in the owner's personal profits, but because only*

such an establishment can pay the working man the wages which his standard of living demands.

The objections of the working men are naturally grounded in their personal experience and interest. The criticisms of the "friends of labor," however, are in most cases disinterested; though it must be said that in too many cases they are based upon a faint acquaintance with the facts. So far as they are disinterested, they must be recognized and met. It is sheer folly to adopt the attitude of the "hard-headed business man" and display a lofty contempt for the increasing interest in the welfare of the working men, which has been developing into a powerful force during the last century. "Hard-headed" is too often a mere euphemism for hard-hearted. Intelligent humanitarianism is not only legitimate but is one evidence of upward progress, and is neither to be ignored nor treated with contempt.

But business men have a right to ask the humanitarians that they be intelligent and informed. It is the business of business men to supply the information needed from the data which are usually at their exclusive disposal. When, for instance, it is alleged that scientific management is "dehumanizing," the charge should be met with the actual histories of men who have worked under this system. Many plants where it has been developed can show, on the part of their employees, an increase of leisure, of interest in their work, of knowledge and improvement in general character, and an enhancement of all-round welfare. Mr. Taylor, and especially Mr. Gantt, have not ignored this side of the subject; but, in proportion to the mass of data in their hands, their contributions have not been full enough. This committee must have had an opportunity to look into this side of the case; and it is to be regretted that they have not improved it more fully.

In a remarkable book, Fatigue and Efficiency, by Josephine Goldmark, published by the Russell Sage Foundation, a chapter is devoted to The New Science of Management: Its Relation to Human Energies. In this chapter are pointed out the advantages of the new methods from the point of view of the working man. After enumerating the perversions to which scientific management may possibly be put, the author says: "If the unscrupulous use of scientific management were all that could be charged against it, the system could defend itself easily enough. . . . More serious is the contention that the effi-

ciency engineers themselves have failed to gauge fairly the tax of increased productivity upon the workers. . . . What we need as regards both men and women (and the only answer which will allay the suspicions aroused by scientific management) is more knowledge as to the ultimate physical adjustment of the workers to the heightened intensity of their tasks." And lest this seem to have too philanthropic a sound, I hasten to add the word of a prominent manufacturer, William C. Redfield, of Brooklyn. He says: "Once for all, let it be said that no management is scientific or permanently profitable which either promotes or permits human overstrain, or which taxes the future of women and children." This reflects, I think accurately, the judgment of many thinking people. The problem of the health of employees is one which demands and must have adequate consideration. No one who knows scientific management at first hand can deny that it has been given such consideration by those who are entitled to class themselves as scientific managers. Miss Edith Wyatt's investigation of the subject with reference to women workers, the results of which are described in Chap. 7 of the book by Clark and Wyatt, "Making Both Ends Meet," is conclusive on this score (and incidentally is somewhat misrepresented in Miss Goldmark's discussion). Numerous instances of the effect of this work on the health of employees might have been collected by the committee and should be given.

I know of a case of a girl who asked to be put on a task 50 per cent greater than what she had theretofore performed. At the time she appeared to be in a sickly condition. The factory nurse was consulted as to the advisability of allowing her to undertake the task and gave her consent on the condition that the girl be allowed to go back to her ordinary work if at the end of a fair period it was evident that the task was too severe for her. After the expiration of four weeks the nurse reported (and it was already evident to the manager) that the girl was not only doing the task easily but had greatly improved in health. This in the nurse's opinion, was due partly to the improved condition under which the girl worked, to the better method she had been taught, to the higher wages she received, and to her increased contentment. This same girl shortly afterward asked for an increase of 33 1/3 per cent more in her task, but this was refused.

In this report the committee emphasize the "transference of

skill " as the basic feature of the new labor-saving management. Unfortunately, however, it appears that this term is used with two meanings. Throughout most of the report it seems to mean the accumulation of skill by the planning department and its transference from this department by actual instruction to the workmen just as machinery is said to be the transference of skill, according to the report, from the designer and draftsman to the machine. The idea intended to be conveyed is undoubtedly right, but the illustration chosen is unfortunate.

Transference of skill, when considered with reference to the industrial revolution and the introduction of machinery, might easily be interpreted to mean the transference of skill from the workman to the machine. What actually happened was that the machine brought to the aid of the workmen some of the vast forces of nature; and in addition it superseded the skill of the hand worker. That this was a distinct loss to the hand worker who was unable to adjust himself to the new conditions is incontestable in the face of economic history; and the sad record of the change is a solemn warning to present day managers to take every step possible to make the adjustment to new methods of management as easy and gradual as possible. The machine developed a new kind of skill on the part of the operator; but now it is a minute skill easily acquired and subject to sudden loss with the change in the design of the machine.

The old-fashioned, all-round workman has disappeared or is rapidly disappearing; and though his replacement by the modern, keen-eyed, high-strung, quick-moving specialist is not to be altogether deplored, it has had certain serious consequences. It has made the present day operator narrower in his knowledge of industry and his skill less adaptable and elastic. It has also rendered obsolete the old methods of apprenticeship, and the present chaotic condition of this subject presents a striking illustration of the failure of managerial thought to bring about the necessary readjustments.

H. M. Wilcox[1] wrote that he felt that through the unfortunate adoption of the term labor-saving management, the committee have allowed themselves to lose the perspective of the subject of scientific management as a whole. The definition of the word science is " knowledge duly arranged and systematized, and this

[1] Miller, Franklin & Co., 17 Battery Pl., New York

so-called new element in industry is chiefly the arranging, systematizing and recording of principles evolved by the art of management, so that the term scientific management is the most logical to use. The term labor-saving appears to have detracted from other phases of scientific management.

In a conversation with Mr. Frank O. Wells, his attention had been called to the fact that practically every industrial plant of any size had a purchasing agency for materials, but how many had a purchasing agent for labor?

H. L. Gantt. The difficulty of making a satisfactory report on the present state of the art of industrial management can be thoroughly understood only by those engaged in the installation of the new methods. The Committee have caught fully the present spirit of the movement now in progress, and Pars. 45 to 58 of their report seem to me to be an excellent resumé of the subject.

In Par. 21, however, we find the following statement: "Another tendency, less pronounced in character, has as its object the improvement of the personal relations between employe and employe, and between employe and employer."

This tendency which is described as being less pronounced, I believe to be the most important part of the whole subject, for until proper relations are established between employer and employe, no system of management or training can be permanently successful.

When the methods described by the committee were first presented, many people thought that they were simply new schemes for exploiting the employe for the benefit of the employer. However false this impression may have been, it was undoubtedly widespread, and formed a serious obstacle to their introduction. How deeply rooted this idea was in the public mind can best be appreciated by reading the discussions of my paper on The Training of Workmen.[1]

Since that time the prominent writers on the subject of management have emphasized the necessity of establishing proper relations between employer and employe, realizing that no scheme that does not recognize this as an integral part can be permanent. Unfortunately this has not yet been acknowledged by everybody to be a fact. Many people value these methods

[1] Trans. Am. Soc. M. E., vol. 30, p. 1053.

only as new ways of controlling workmen for their exclusive benefit, and become interested in them only in times of trouble. Financiers seem particularly prone to take this view.

To illustrate this point, about a year ago I was consulted by a large corporation, which, however, took no further steps. Recently the same corporation was much interested in the subject of management, and I found that its employes were on the verge of a strike. The strike, however, did not come off, and the interest of the management apparently died out when the danger disappeared.

While the idea of exploiting workmen for some one's benefit is obnoxious to most people engaged directly in industrial pursuits, this is not the case with those farther removed from the workmen.

If I am to judge from the letters I received after the hearing before the Interstate Commerce Commission in Washington in November, 1910, many people thought they saw in this movement a chance to get something for nothing. So strongly was I impressed with the extent of this attitude that I felt impelled to condemn it most strongly in my address at Dartmouth a year ago. When I was leaving the hall after that address, a benevolent looking old gentleman came up and said: "Of course you will modify that before it is published?" and I did, but not just as he meant.

I regret that the committee did not include in their bibliography of this subject, books and papers published elsewhere than in the annals of the Society, for some of the very best work on this subject has been done in the last three years, during which time the Society has nothing to show. I wish to call especial attention to the work of Major Hine on Modern Organization and to that of Hon. William C. Redfield on The New Industrial Day. Both of those books, just published, will well repay careful study, as they are in accord with the democratic as opposed to the autocratic spirit in industry.

With regard to the minority report, a careful reading seems to indicate that the writer of it is practically in accord with the majority, except that he wishes to take a shot at the statement made before the Interstate Commerce Commission that the railroads were losing one million dollars per day. I must confess that when this statement was made it seemed to me rather extravagant, but a careful study of conditions for the past two

years has convinced me that the statement was conservative rather than otherwise if all preventable losses are included.

The writer of the minority report is connected with what I believe to be one of the most progressive and best managed railroads on this continent. I did a little work for him a few years ago, first, having in conjunction with Messrs. Dodge and Day of Philadelphia, made a report on the conditions in his shops. Our report expressed the opinion that his method of operating these shops was at least as good as that of the best railroad shops we knew anything about, and we visited several representative shops in this connection.

The real inefficiencies with which I came in contact on this road, however, were not in the shops. They lay in antique time-keeping, record-keeping, cost-keeping, and purchasing systems, which were run, not for the benefit of the maintenance and operating departments, but apparently to hamper these departments to the greatest possible extent. Inasmuch as most roads handle these functions in substantially the same manner, I believe that the inefficiencies of railroad operation, due to the lack of appreciation by the financial end of the needs of the operating end, are far greater than those over which the operating end has absolute control.

Railroads are built nominally to earn money for their stockholders by the sale of transportation. If this were the only way they could be used to get money, all activities connected with a railroad would be so run as to assist in the economical production of transportation, just as all the activities of a well run foundry are harmonized for the production of castings at low cost.

As a matter of fact, the financial end, including the purchasing, store-keeping, time, cost, and record keeping, does not as a rule consider economical operation as any of its affair, but puts this responsibility entirely up to the operating force. The best railroad managers of today, however, see that real economical operation cannot be had unless all functions contribute to that end, just as they do in the best managed factories.

Unfortunately this is a view that the average financier finds difficulty in accepting, especially if he has always been a financier. A proper comprehension of this subject, however, is the best way to head off the growing sentiment for the government ownership of railroads.

Turning again to Par. 21, concerning the relations between

employer and employe, any scheme for the training of workmen, or promoting the transference of skill, if that term is preferred, to be ultimately successful must carry with it a guarantee on the part of the management that the employe shall receive his share of the product of his increased efficiency or skill. Increasing the efficiency of a workman thus differs radically from increasing that of a machine, which claims no share of the results.

If, however, the attempt to increase the efficiency of the workman is made in such a manner as to lead both employer and employe to feel that he is getting what he should from their mutual efforts, the promotion of efficiency is sure to follow, and we have a working basis on which to attempt the solution of some of our industrial problems. As long, however, as such prominent men as Mr. Parry and Mr. Kirby of the National Association of Manufacturers continue to use such language as they have in the past in speaking of workmen, just so long will the workmen continue to use brute force in dealing with employers, and a satisfactory solution of the labor problem be delayed.

The implication in the report of the committee that the development of methods and the training of workmen are functions of the management has to my mind not been sufficiently emphasized. The piece-work system as usually operated, and most of the premium and bonus systems in general use do not recognize training as a function of management, and may be classed as individual effort systems. They have undoubtedly accomplished some results in the past, but we have now reached the stage in our complicated industries where more than individual effort is required. Individual effort may be so hampered by environment as to make progress impossible, and to cause the workman to become so discouraged as to give up whatever effort he may be making. On the other hand, if the management determines that a certain degree of efficiency is attainable, accepts the responsibility of training workmen up to the point necessary to obtain that efficiency, and provides means for obtaining it, we have an entirely different condition.

This view of the subject is so opposed to that held by most managers, superintendents and foremen that it is with great difficulty that we can get them to accept it, and the time needed to bring them around to this viewpoint is far greater than most realize. There are many reasons for this, and the fact that a

man does not instantly accept a new proposition, which is to his mind revolutionary, should not for one minute be held up against him, for a strong man who has been successful in following one line of action, justly regards an entirely different line as being inferior to one which has brought him his success.

Fig. 1 represents the work of women winding yarn in a cotton mill after the system had been in effect one month, at which time three operatives were regularly earning a bonus. At the start, one of the best workers was offered the opportunity to perform the task and to earn bonus, but she felt the task was too severe and did not wish to try. A week later, however, she did begin the work and made her bonus right along. Some of the people in this room were young girls who apparently had but little interest in the work and made practically no effort to earn their bonus. The older and more staid ones, however, took hold promptly and the younger ones gradually dropped out. The newcomers at first spent three or four weeks before they actually earned the bonus. As, however, those who were not interested in earning the bonus gradually disappeared, the habit or fashion in that room came to be—attend to your job and earn the bonus. The newcomers then in many cases earned their bonus from the very first day. This chart is dated March 1909. Fig 2 shows a chart of this room as it exists today and is reproduced by permission of the treasurer of the mill.

Some months ago the Industrial Workers of the World made an attempt to cause a strike in all of the mills in the town where this mill is situated, and actually succeeded in shutting down some of the mills for several weeks. In this mill, on which a strong attack was made, only 60 out of 600 employes were drawn out, and the management had no difficulty whatever in filling all their places in a few days.

Fig. 3 represents the work of girls in a worsted mill. The best workers were put on bonus work first. Their improvement is noticeable. It was nearly three months before all were put on this work. The poorer ones failed at first, but on the last day shown on the chart all of the poorer workers earned bonus. During the period from April 15 to July 10, one-hundred and sixty-one girls had been put on bonus work, and 21 out of this number had for one reason or other left the employ of the company. The reasons were as follows: four left on account of poor health or were dissatisfied; one married; one entered a convent; one went

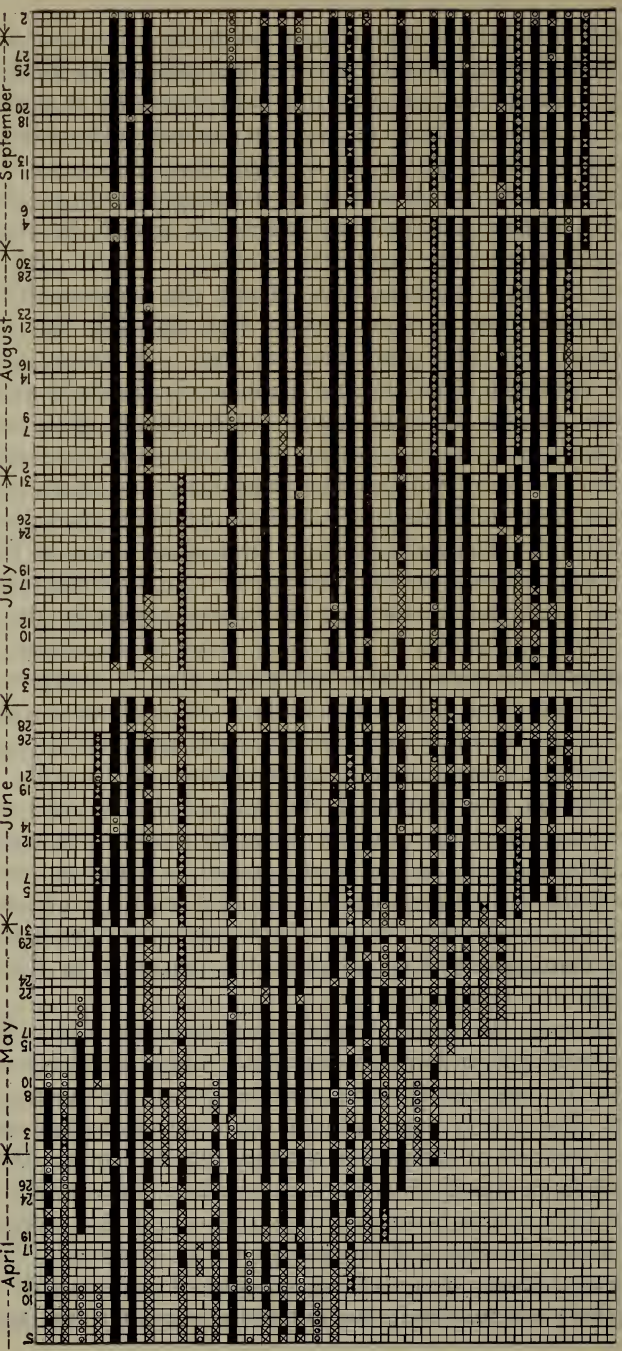


FIG. 1 CHART SHOWING RESULTS OF BONUS SYSTEM: WOMEN WINDING YARN IN COTTON MILL, 1909

Each line is the record for one worker. Note how much more quickly new workers put on in June began to earn bonus than did those in April or May

to work in the mountains for the summer; five left town and nine were discharged.

One of the obstacles nearly always encountered is the opposition of certain people in authority who, having once expressed their disapproval of these methods, feel it their duty to oppose

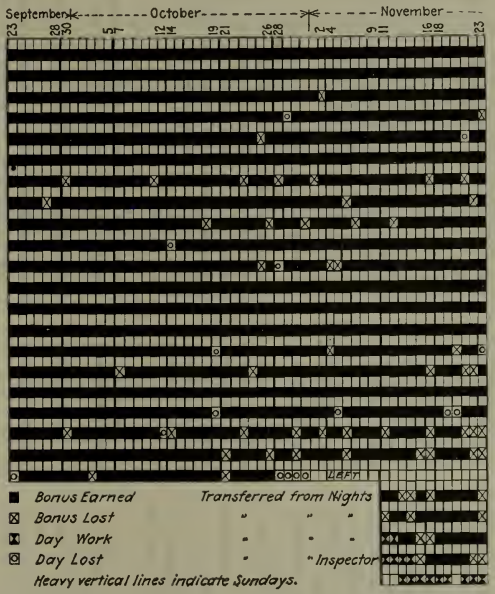


FIG. 2 CONDITIONS EXISTING IN 1913 IN THE DEPARTMENT REPRESENTED IN FIG. 1

their introduction to as great an extent as possible. Their original arguments having failed, the argument that we were overworking the girls was advanced and they insisted on being allowed to put on piecework some of this work which was in another building, and which had not yet been put on the task and bonus system. Accordingly, on June 19 their piecework was started. On July 11, the date of the report, 19 out of a total of 53 girls in this building had left. In other words, almost one-third of the girls who were put on piecework according to the request of the man who had their interests most at heart, left in three weeks. In the case of our bonus workers, approximately one-eighth of the girls left in 15 weeks.

In 1904 I put this system into a shop making packing boxes. A few weeks ago I had word that it was running substantially

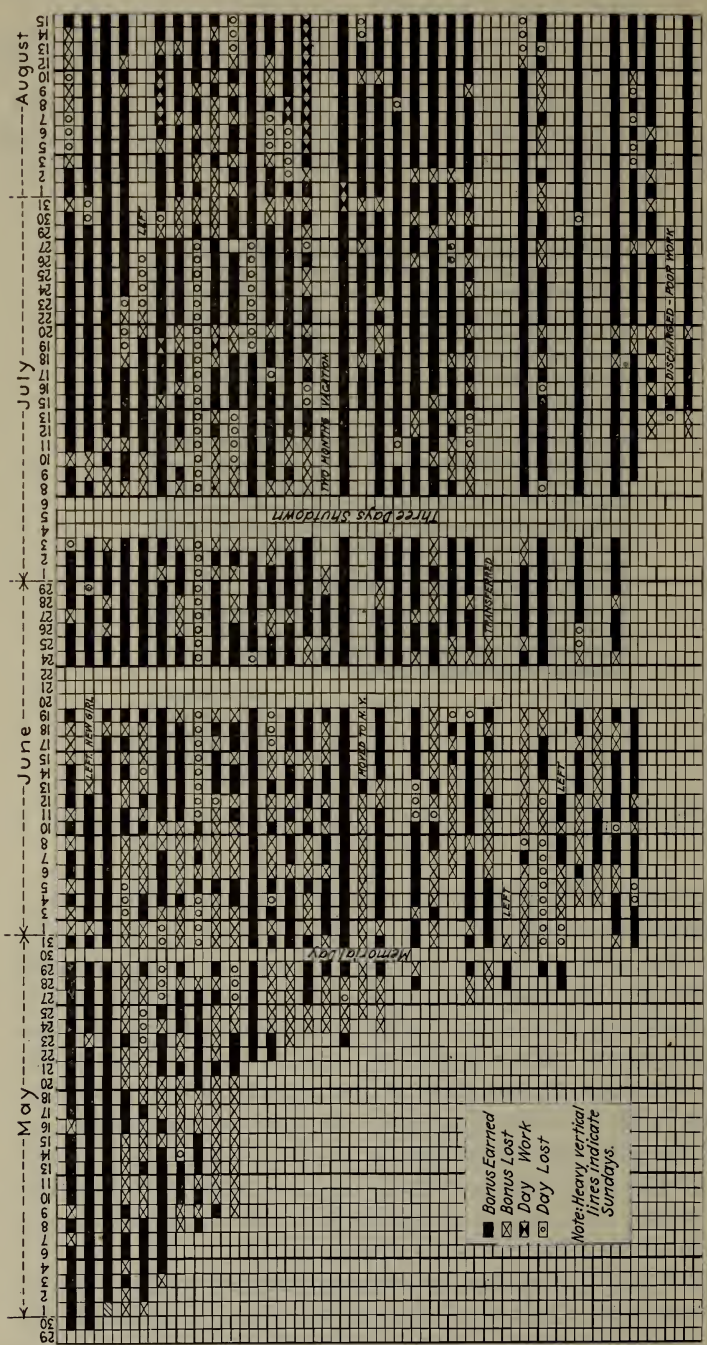


FIG. 3 RECORD OF WORK IN WORSTED MILL

Each line represents one worker. This is a sample chart for one room. There were several other rooms

as it was started and that nearly all of the original bonus workers were still there.

Fig. 4 represents the making of sheets and pillow cases and the work of starting the task and bonus was done by a man who had been connected with me, but who was doing this on his own responsibility. I was not personally in touch with this work when it was done. The factory was shut down on a number of days, November 28 (evidently Thanksgiving day) and then on Christmas and all Wednesdays and Saturdays for the next two weeks. The work started off very well, but the rush to get people on bonus on November 30 evidently upset things, for a number of workers are back on day work. This was probably due to the inability of the task setter to set tasks on new work fast enough. Just before Christmas week the same condition obtained, and after Christmas there was not enough work to keep the factory running full. However, by the middle of January those who had bonus work were beginning to earn their bonus pretty regularly, and by February 10 the number of workers was just about large enough so that each could be supplied with a full amount of work. From that time on the work went smoothly.

It is evident from this chart that the management has its part to play in supplying work and teaching workers, and that the trouble was not with the workers but with lack of proper balance in the managing department. The fault, however, is a natural one, for when the man responsible for the output finds the advantage he can obtain by the task and bonus system he almost invariably insists on putting as many people on bonus as possible, with the result that he finds he cannot supply all the workers properly and that numbers of them have to be put back on day work.

You will ask why the task setter does not explain this to the superintendent and make it clear that that is the wrong way to do. I can only say that no amount of explanation on my part, or that of my representatives, seems to have much effect and we have about come to the conclusion that the best way to do is to let them make their mistakes and find out, then the question is settled once and for all time.

Fig. 5 is a particularly interesting chart inasmuch as it is the most recent—the development is still going on. It is reproduced by the courtesy of the manager. In this the work went

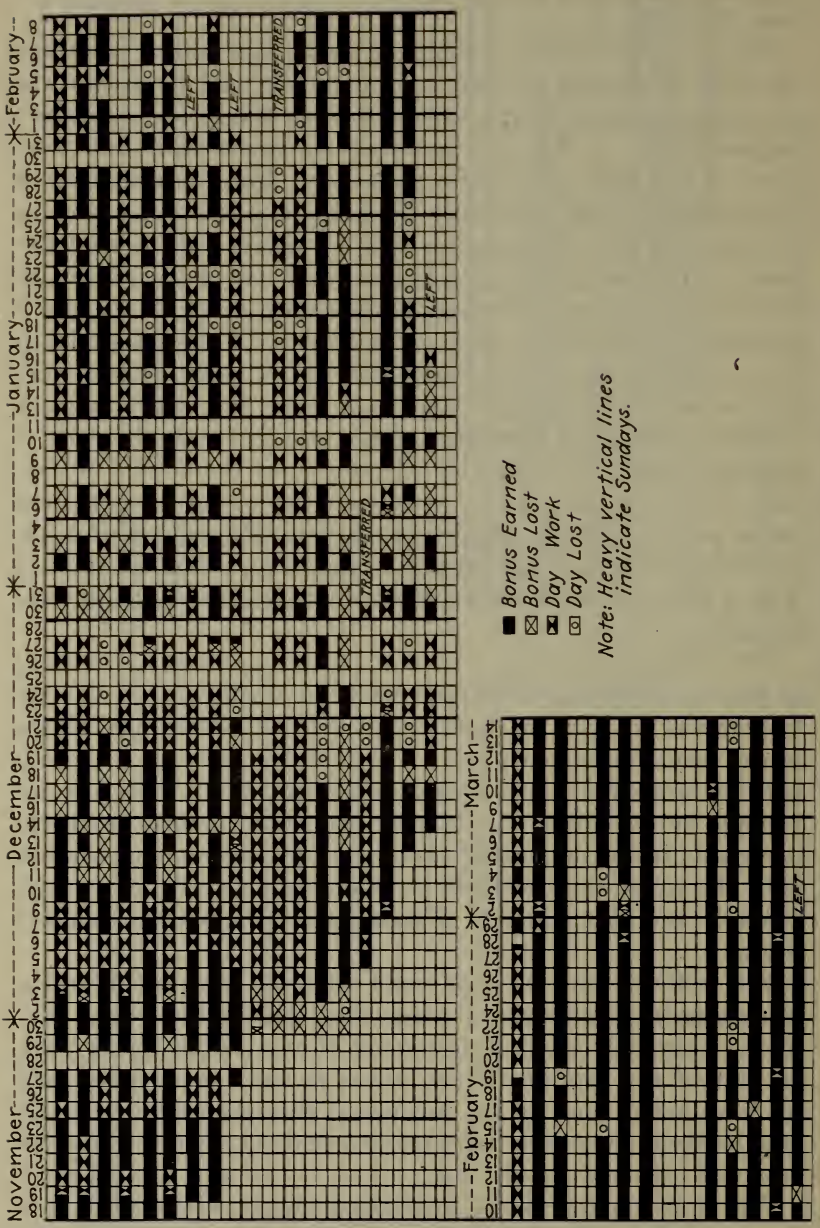


FIG. 4 RECORD OF WORK IN MILL MAKING SHEETS AND PILLOW CASES

very well until we began to push it too fast. When the desire to get an increased output overcame the conservatism with which it is necessary to establish a new method, those workers who were last put on influenced the others not to perform the task, and on one day nobody made bonus. One girl, however, felt that she needed the money and continued with the work. The others showed their hostility toward her in a number of ways, but she still persisted. After they had ceased to try to do the work for three or four days, several of the girls sent in their notice that they were going to quit. They were evidently trying to raise an issue, but inasmuch as the management ignored it and went about its business of getting conditions so perfect that the workers could have no cause whatever for not trying to earn their bonus, they were unable to get the issue accepted. At the end of eight days some of the girls became willing to try again, and from that time on the number increased. For some time past the applications from other girls to work on task and bonus have been so numerous that we have been unable to provide proper conditions and to set tasks for them fast enough.

The question naturally arises with respect to the charts as to the ratio between the amount which is being accomplished now and what was accomplished previous to starting in with this work. In order to make these comparisons clear we devised what is called a Percentage Chart. On this the average production previous to the installation of our methods is called 100 per cent. Our new production may be two, three or four times that amount. The wages paid before the work was started are also rated at 100 per cent. The new wages would be an increase over the old standard, usually of from 30 to 50 per cent and the wage cost measured in the same manner would be distinctly below the previous wage cost.

Fig. 6 shows these ratios with regard to work done several years ago in a bleachery in Rhode Island. Each of the vertical lines represents a different kind of work. The horizontal black line marked 100 per cent represents the amount of work which was done on each of these operations previous to our investigations. The upper lines represent the amount of work now being done, compared with what was previously done. The heavy black line marked 100 per cent also represents the previous wages paid, and the dotted lines above it represent wages now paid. The 100 per cent line also represents the previous wage cost. The

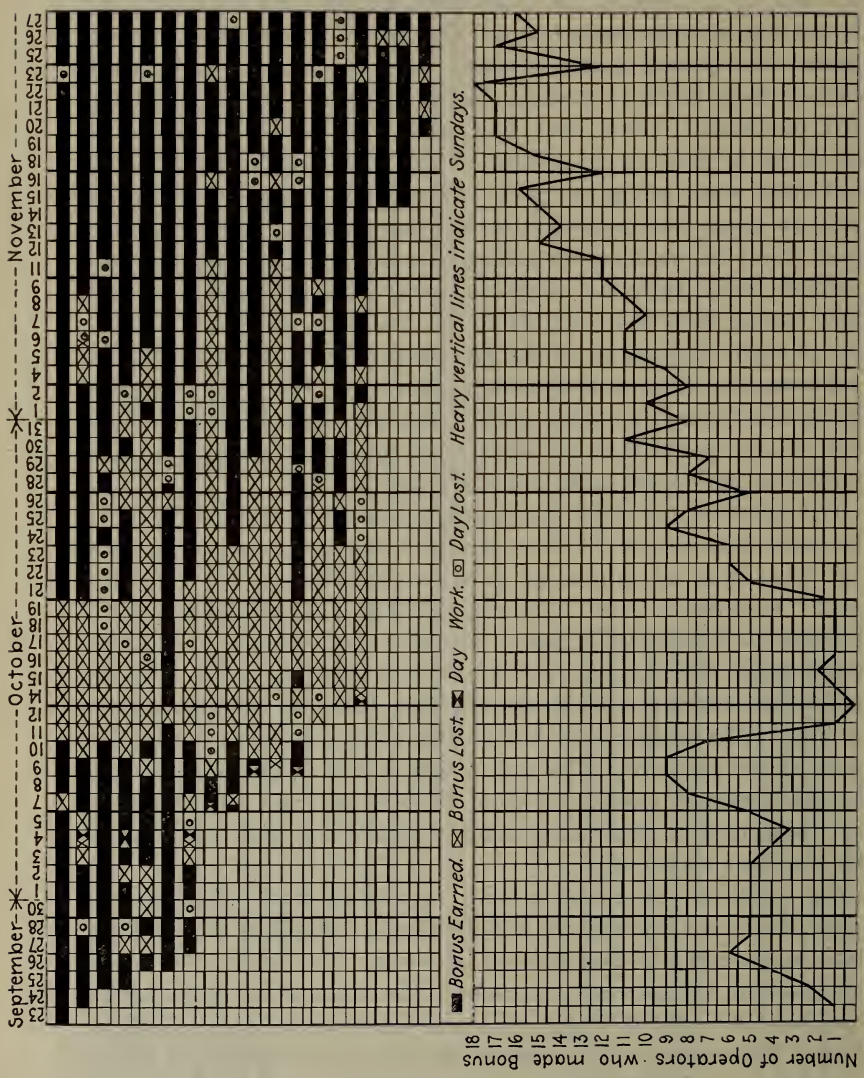


FIG. 5 CHART ILLUSTRATING BONUS WORK IN FACTORY WHERE THE EFFORT WAS MADE TO INCREASE THE OUTPUT TOO RAPIDLY

dotted line below represents the present wage cost. The increase in product is around 200 per cent and the decrease in wage cost is approximately 40 per cent, while the increase in wages is also 40 per cent. It should be borne in mind that *this increase in product is not due solely to the work of the operative, but also to more careful study and coöperation on the part of the management as well.*

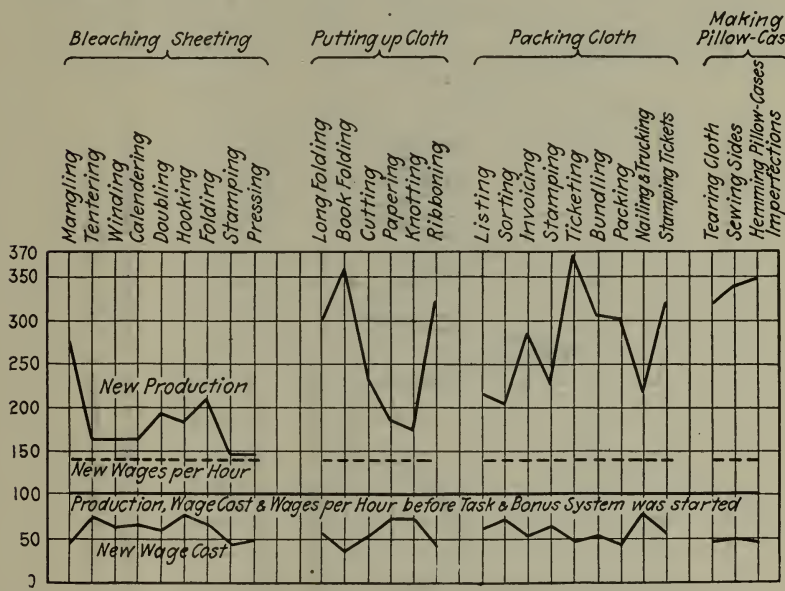


FIG. 6 ACTUAL RESULTS UNDER THE TASK AND BONUS SYSTEM AFTER MORE THAN TWO YEARS' OPERATION

Each Vertical Line Represents a Different Operation

Fig. 7 shows similar results for work on small automatic screw machines. In this case the light line represents the task; the upper heavy line the amount of work done; the upper dashed line the wages now being received; the lower dotted line the new wage cost. In this particular case the shop was well run before we undertook to study the work, and the workmen were getting good wages. The increase in production is not quite so high as in Fig. 8 which shows similar results for large automatic machines. Fig. 9 represents the same change for miscellaneous machine work in a plant manufacturing a small article in quantities.

Comparing all four of these charts it will be noted that there

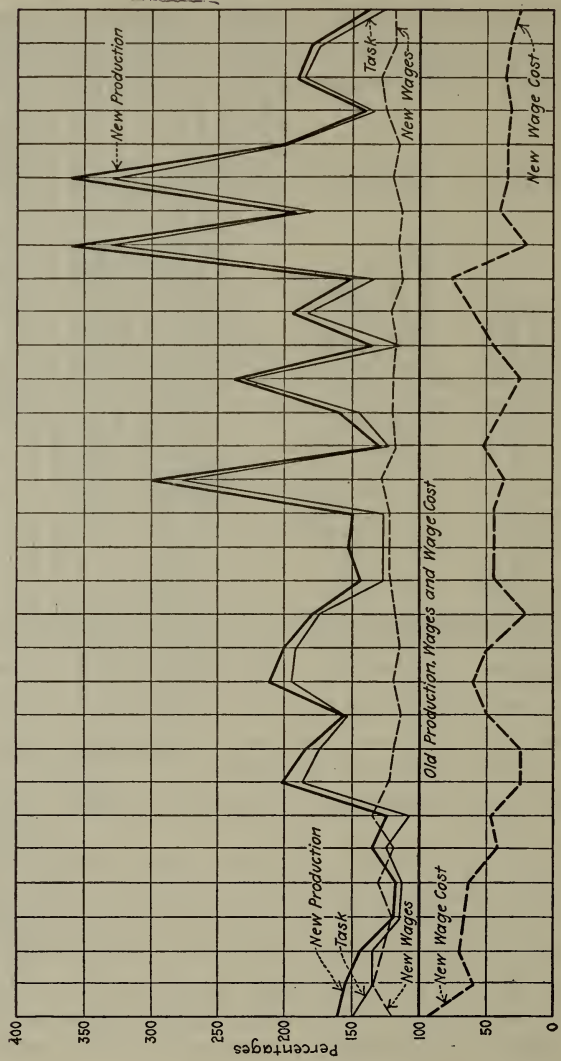


FIG. 7 WAGES AND PRODUCTION DIAGRAM ON AUTOMATIC SCREW MACHINE WORK
Each Vertical Line Represents a Different Operation

is a very striking similarity whether doing handwork in a bleachery or automatic lathe work in a machine shop. If the management assumes its share of responsibility in preparing the work, in seeing that the machines are in proper condition, and in training the workmen, we can get from two to three times as much work done as is usually done, pay 20 to 50 per cent increase in wages and still save about 40 per cent in wage cost.

This increase in output brings down the overhead expense on every unit of product so that the decrease in wage cost is not the only important item. It is not even the most important.

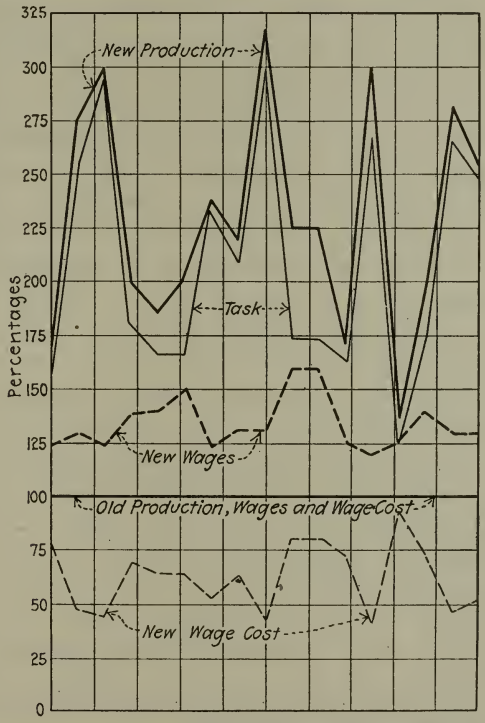


FIG. 8 WAGES AND PRODUCTION DIAGRAM ON LARGE AUTOMATIC WORK
Each Vertical Line Represents a Different Operation

Unless the overhead expense is markedly increased when the product is increased this expense per unit of product comes down substantially in inverse ratio to the amount that the product goes up. The reduction in cost from this source is usually markedly greater in dollars and cents than the reduction in

wage cost. This side of the cost question has been given too little consideration. Andrew Carnegie was among the first men in this country to recognize the great value of getting a larger product from his plants, and this, perhaps more than any other fact, gave him the mastery of the steel business. Many times we can afford to pay even a higher wage per piece if thereby we can reduce this overhead expense. In general, however, a thorough study of the work enables us to reduce both wage cost and overhead expense per unit of product at the same time substantially increasing the earnings of the workman.

In Fig. 9 will be noted a discrepancy in certain cases between the task set and the amount of work performed, showing that the workman did a great deal more than expected. It is asked, naturally, how we could overcome this difficulty, for many would feel at once that a serious mistake had been made and that the tasks should be increased or the rates reduced.

In reply, these tasks were set by a task-setter who had not had sufficient experience. We, however, do not consider that because he has made an error that it is necessary for us to change the rates. As a matter of fact, we rather prefer that there should be a few easy tasks so that the workmen may have a practical demonstration of the fact that we are not going to cut rates and that they need have no fear whatever if they do all the work they can and earn all the money possible.

Another comment may be that these shops must certainly have been run very badly before. While some of them perhaps were not in the class of well run shops, others were, and high up in the class. Before the introduction of these methods the results achieved were due to the effort the workman put into his work, with practically but little direct assistance from those over him. After the methods were installed he was taught the best way that we could devise of doing the work, offered a substantial reward for accomplishing the desired results in the manner in which he had been taught, and the conditions under which he was working were so modified that these results could be accomplished if the worker had been properly trained. In other words, these results are not accomplished by the workman alone and unaided. He had the thorough coöperation of a strong management.

Another criticism may be that no account is taken of the indirect labor, such as transportation, clerical work, etc. In reply,

we find as a rule that there has been even less attention given to the proper study and planning of indirect labor than to direct labor, and that the chance for improvement in that line is quite as great as, if not greater than, in the line of direct labor.

My object in presenting so many charts which illustrate the same thing, is to show to as great an extent as possible that the principles laid down in this discussion are of wide application;

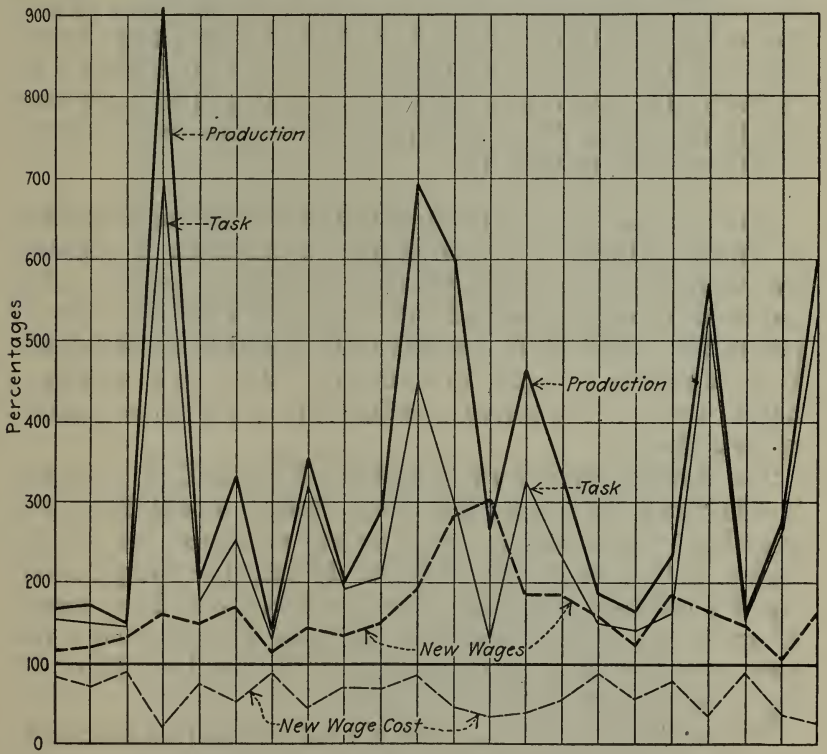


FIG. 9 WAGES AND PRODUCTION DIAGRAM ON SMALL MISCELLANEOUS MACHINE SHOP WORK

Each Vertical Line Represents a Different Operation

and that in all cases substantial results may be expected if these principles are properly carried out.

The essential point in carrying out these ideas properly is that they should be *understood, and thoroughly appreciated by the people who undertake their application.* The results shown cannot be accomplished unless we have harmonious coöperation

between employer and employe, which is impossible under the form of management which assumes that the responsibility of the manager has ended when he has issued his instructions to the shop.

To my mind, the training of workmen to fill all the different positions in a factory is one of the important functions of the management, and we all know that training is slow and expensive, but it is the only method which holds out any hope of producing even a partial solution of our present industrial problems. This training cannot be accomplished by following blindly a lot of rules. The men responsible for the training must understand clearly the reasons for every step before they can teach employes to follow them intelligently.

JOHN G. ALDRICH. I believe that the report of this Committee is right. Scientific or labor-saving management is scientific measurement and everyone will agree that this is desirable. The importance has been pointed out in this report of the study of production by scientific methods and of setting up standards. It is impossible to make up standards without first making a careful study of the correct motions and of the times required to make them.

The New England Butt Company, of which I am manager, is doing some work along this line to which I would like to call attention. This company builds largely machinery for making braids, such as trimmings for ladies' dresses, shoe strings, coverings for insulated wires, etc. The machines are made up mostly of small light castings which are machined but little, but which must be well made so that the parts will fit together without filing or other hand work.

Within the last few years with the help of various experts we have continually made improvements in our manufacturing departments. Time study was made of different operations and of different methods and proper times were set on those methods upon which to base the various systems of payments for work performed. We are now using a method of time and motion study which has not been used before, and which not only gives more accurate results than have been heretofore possible, but also enables us to discover methods that are much more economical.

This method consists of taking motion pictures of the various operations with a special moving picture camera, and photo-

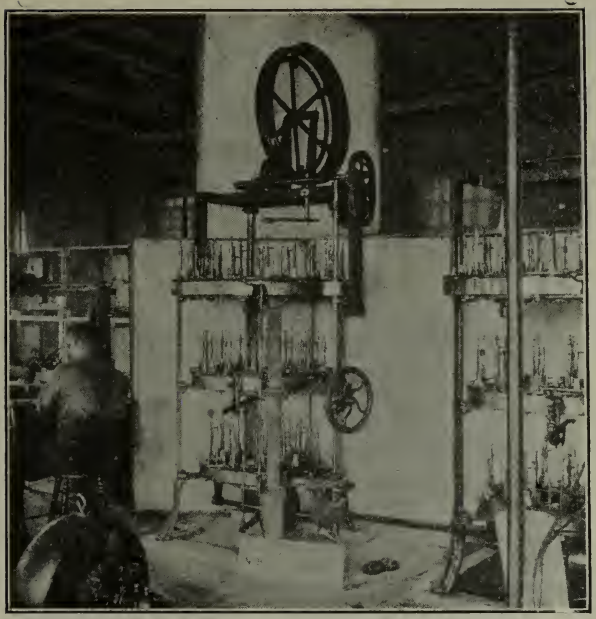


FIG. 10 DOUBLE DECK BRAIDING MACHINE

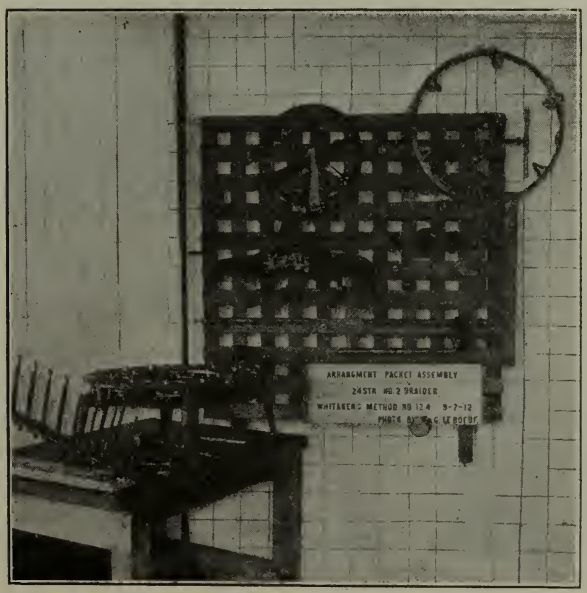


FIG. 11 TYPICAL PORTABLE ASSEMBLY BENCH AGAINST PACKET WITH THE ARTS ARRANGED FOR ASSEMBLING A BRAIDING MACHINE

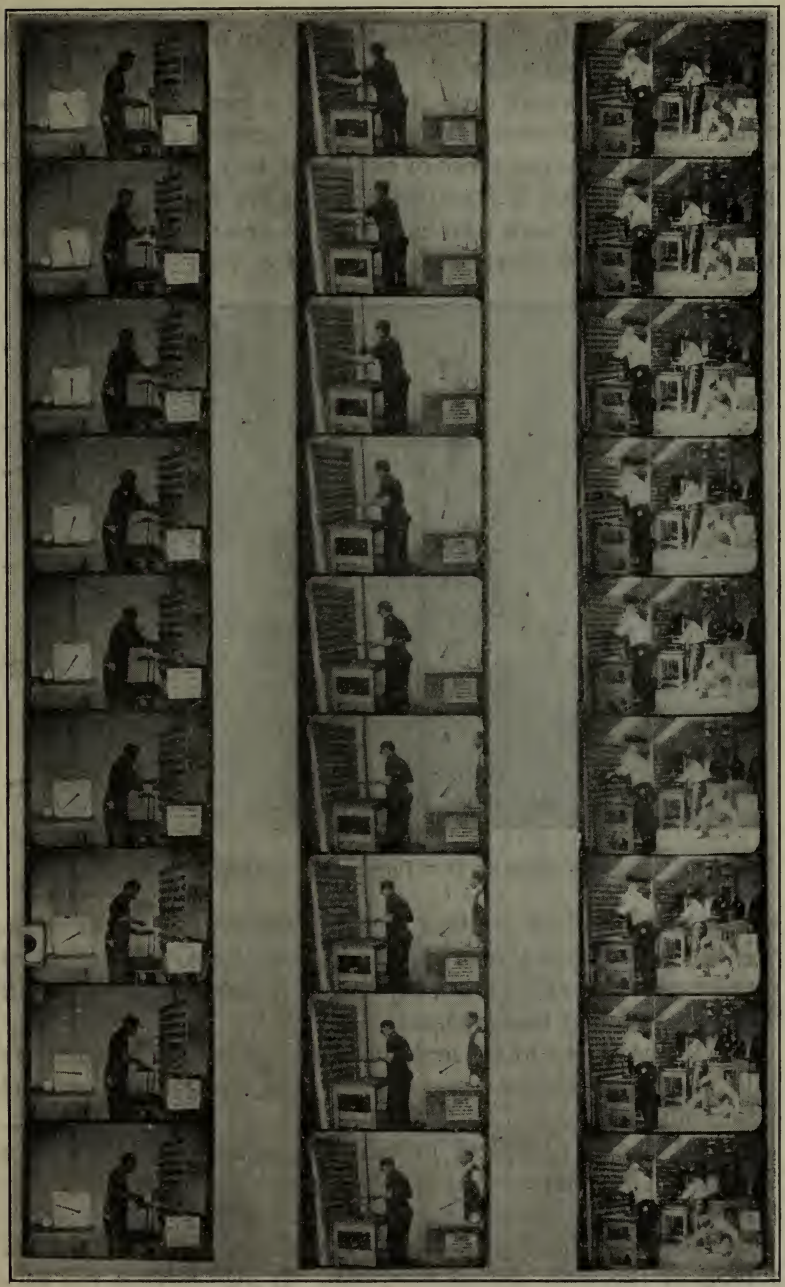
graphing in each picture a clock of special design showing minute divisions of time. The hand of this clock revolves once in six seconds so that the divisions represent thousandths of a minute and are easily read to half thousandths. The continuous motion picture film furnishes permanent record of times and motions. To develop improved methods this film is afterwards studied with a magnifying glass and it is not necessary to project the pictures on a screen.

Before taking these pictures considerable study was given to-

FIG. 12 MOTION PICTURE MACHINE RECORDING THE MOTIONS OF A WORKMAN

ward eliminating waste motions and otherwise improving the conditions under which the work was done. In assembling machines, instead of picking up the pieces to be assembled from various boxes, packets were arranged with the parts placed in convenient positions, and also placed in the proper sequence, so that no mental process was required of the workman to determine or select the parts to come next.

Since using this method, which we have designated micro-motion study, previous times have been reduced over two-thirds. Its records have suggested to us methods that now permit in one



FIGS. 13, 14, 15 SECTIONS OF MOTION PICTURE FILMS OF WORKMEN ASSEMBLING A BRAIDING MACHINE

case doing work in 8½ minutes that before using micro-motion study took 37½ minutes.

Par. 6 of the report calls attention to a most important principle, " the transference of skill." Micro-motion study furnishes a means for the transference of skill from man to machine. More important than this it furnishes a means for the transference of experience from a man who has had it to one who has never had it. We have used micro-motion study for determining the cor-

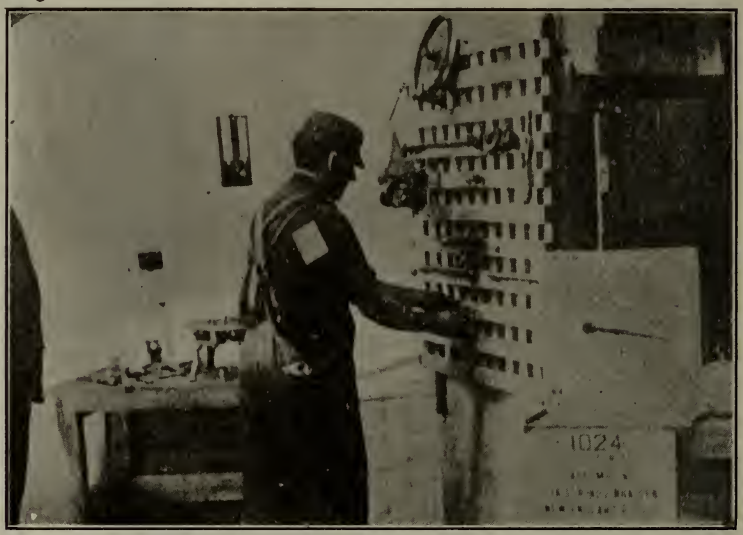


FIG. 16 ENLARGEMENT OF ONE PICTURE OF A MOTION PICTURE FILM

rect times of the best motions in many different kinds of work. It is the least expensive as well as the only accurate method of recording motion and time study data. A few illustrations will give a clear idea of this process:

Fig. 10 shows one of the styles of braiding machines which we build and to which we have applied this method of time and motion study.

Fig. 11 shows a typical portable assembly bench against a packet with the parts arranged for assembling a braiding machine with the fewest and shortest motions.

Fig. 12 shows the motion picture machine recording the actual motions of the most skilled workman. It also records the subdivisions of each elementary motion. It will be noticed that the floor and background are both marked off into 4-in. squares.

The object of this is to enable us to place the bench in the most desirable location and to see that the workman stands in the most convenient position. Furthermore, it enables us to determine the lengths of all motions.

Figs. 13 and 14 show sections of two typical motion picture films of a workman assembling a braiding machine. The clock can be easily seen in these pictures and by means of it the times of each elementary motion can be determined. In the consecutive views the clock shows elapsed times of three one-thousandths of a minute.

Fig. 15 shows a section of a motion picture film with several workmen assembling the same kind of machines. It shows how different operations are recorded at the same time for future study, thus reducing the cost of time study. It also shows how the performance of a number of men on the same kind of work can be compared.

Fig. 16 shows an enlargement of one of the pictures of motion picture films.

From the above it will be realized that micro-motion study enables us

a To capture the experience of the most skilled workman and record it for the benefit of all.

b To determine the motions of least waste.

c To teach the best known methods only.

d To analyze, measure and compare new data so that improved methods may be constantly standardized as fast as discovered.

I believe that the time will soon come when we will have a national bureau of standards of best methods, and micro-motion study will provide a means that the government can use for collecting and recording the best practice of the workmen in our industries.

Robert Thurston Kent. Underlying the three regulative principles given in Par. 41, is a greater one which renders them all effective—the principle of measurement. Micro-motion study is the most powerful tool ever offered the engineer to measure the efficiency of the worker. The moving picture machine together with a specially constructed clock which will measure time intervals to the half-thousandth of a minute, presents a means of studying operations or cycles of operations which occur

too rapidly for the eye to follow and which, therefore, are impossible of study by the ordinary method of an observer with a stop watch.

Time study is the basis of all modern management. The provision of a machine to make time study should be as revolutionary in the art of time study as was the invention of the power loom in the art of weaving. Among other things it absolutely eliminates the human equation. It provides a method in which there is not only no possibility of error in measurement, but which furnishes at once a true statement of the time elapsed in the performance of any operation and a record and instruction card of the best method of doing a job.

In The Psychology of Management the following statement appears:

> Measurement is a most necessary adjunct to selecting the workers and the managers and assigning them to the proper functions and work. They cannot be selected to the greatest advantage and set to functionalized work until (*a*) the unit of measurement that will of itself tend to reduce costs has been determined; (*b*) methods of measurements have been determined; (*c*) measurement has been applied, and (*d*) standards for measurement have been derived.

It is obvious that micro-motion study at once presents a means of most economically determining and applying all these standards.

It was the writer's privilege to be present at many of the experiments which led to the development of micro-motion study and to observe how it was applied to the transference of skill and experience. Scientific management makes use of the best methods of all trades which can be transferred to the trade under consideration. The experiments under which micro-motion study was developed are an illustration of this point. The work involved was the assembling of braiding machines, obviously a machine shop proposition. The original method of assembly involved the bringing of the various pieces to the job in a box or boxes from which they were taken as required by the worker. Many unnecessary motions were used to transfer the individual pieces to the growing machine which were promptly revealed by the moving picture machine. A bench which would bring the top of the completed machine at a convenient level for the workman was built and the parts were arranged in an orderly manner in bins behind him, thus reducing the number of motions.

In other trades in which some of the experimenters had been

engaged it had been found advisable to group the units composing a single assembly on a packet arranging them in the order in which they were needed. This suggested a similar packet scheme for the parts of the braider. A horizontal packet was first made up and the motion picture machine revealed an irregularity and lack of rhythm in the movements of assembling which seriously cut down the efficiency of the workman. A change to a vertical packet and a standard height of portable assembly bench was thereby indicated. On this packet the various parts are hung on pegs in that exact position which micro-motion study has revealed as the most economical of motions both as regards time and length of travel. Micro-motion study revealed the deficiencies of previous methods and permitted the development of the final accepted methods in a small fraction of the time and expense which would have been necessary under conditions existing before its invention.

F. A. Waldron. Literature on scientific management has dwelt largely with the labor problem, while the other elements which tend so much toward maintaining factory output have been to a large extent ignored. Scientific management in its highest sense can be likened to the proper functioning of the human body, in which each part carries its own portion of the load, does its own share of the work at the proper time and distributes energy in proper proportions to all parts of the human frame.

There has been more or less tendency in the management of industries to divorce the financial and sales organizations from the factory or shop management in such an arbitrary manner that the two act in a way indicating that each is working for itself regardless of the broader interests of the company of which each is a part. So long as the sales end can get orders to the factory and be duly credited, the fact as to whether this order contains enough specific information to complete it, is ignored and it is left for the factory to make it out as best it can. A serious loss in volume of output may thus be caused. I can recall several instances in my experience where an output of from $50,000 to $100,000 worth of business was held awaiting detailed information on minor items before the order could be shipped. It is essential to keep the entire organization in balance. Do not overload sale and advertising departments if the factory cannot

handle the work. Build up on a solid foundation surely and gradually. Educate not in the factory alone, but along the entire line.

Output. In the excitement of the moment, the quality of output should not be lost sight of. This cannot be maintained by the mere placing of tools, instructions and drawings in the hands of the workman. Men must be trained and it takes time and money to train them. Unless this training embodies the elements of thoroughness and completeness, they become half-rate and slip-shod men, looking for pay day and taking little if any interest in the work.

Ideal factory conditions rest almost entirely on one basic principle which is *constant volume of output.* It is not always possible to maintain this owing to conditions that may arise, such as orders, capital, and the proper supply of materials and help. Every green workman broken in means sacrifice in profits, quality and volume of output. As these weak spots develop, however, we should try to build them up in such a way as to make conditions tending towards constant volume of output, as nearly perfect as possible.

It is far better to work at a steady even gait for a year than to intensify the production to a point where all the work is done in six months and your factory is idle for the remaining six months.

The extent to which time study should be carried depends entirely on these conditions and the expense incurred by ultra-refinement could better be invested, oftentimes, in raw materials or finished stock of proper quality. It is the province of the industrial and efficiency engineer to help guide the manufacturer as to how far he should go into a question of this kind.

A factory with an undersold output is no place for a premium or bonus system. Accurate and simple accounting and routing are requisites here. With an oversold output, a premium, bonus or piece rate system, based on accurate time study, combined with accurate and simple accounting and routing are necessary to produce the maximum efficiency on the shop end, combined with a purchasing, inspection and stock department that can keep pace with a well developed shop organization.

Purchasing. The most effective coöperation between the factory and purchasing departments is as essential as time study or production engineering work. Of what avail is time study

and intensified production if the materials the purchasing agent buys at the lowest price will not meet the requirements of the product in quality? Work will either be thrown away before it is assembled, or if sent out in the finished product, it will react on business. The only way is to buy to specifications and see that the material is on hand before it is needed and that the specifications are conformed to.

Low prices paid for materials are not necessarily exponents of a purchasing agent's efficiency. The man who can discriminate and buy the right thing at a fair price is the kind of man that lasts in this period of competition and exacting requirements.

Serious delays and expenses are often caused by incomplete or incorrect specifications for simple materials and supplies. This trouble can be obviated by a symbol or number system which ties all materials used in the factory to a standard specification which is definite and complete in its wording.

Engineering. A product improperly designed and well made cannot maintain its place in the market, and, conversely, a product properly designed and improperly made, cannot be sold. To design properly and to see that it is properly made, is the measure of efficiency of the engineering department.

Proper strength, lines, proportions, tolerations, inspection, instructions, manufacturing specifications, tool design and manufacturing, should be done in this department and done thoroughly. Time study and bonus system are thrown away if done on work that cannot be used, or hurts trade. Further, responsibility is more readily placed if this is done in the full sense and meaning of the word " complete."

The engineering department in the modern organization of high efficiency is the most important of all, as it is the fountain head from which all specifications are issued, designs made and quality of work determined.

Inspection. To insure proper materials being given to the workmen is the first and most important duty of the inspection department, followed by proper tools, condition of machines, jigs and fixtures for the reasonable performance of acceptable work. These functions are just as necessary to efficient work as time study or bonus system, for without these the intensified production would be lacking, owing to the fact that conditions for perfecting the work would not be constant and the failure

to perfect such work would be beyond the workman's control.

After these requirements have been brought up to the proper standard of efficiency, this division can control the quality of the work as specified by the engineering division and is in a position to insist on its quality being maintained.

Maintenance. To do a full day's work a man must have proper light, heat, power machines, tools and fixtures, etc., in proper working condition, and it is up to the management to see to it that he does. One of the most common troubles is the condition of belts, long ago recognized by Fred. W. Taylor and covered in his paper on Shop Management.[1] His system with the belt, bench and scales in a factory using many belts is the best obtainable and shows direct results in volume of output and saving in belt bills.

Countershafts, main shafts, jack shafts, motors, machine repairs, should receive a systematic supervision to insure constant and efficient operation. It is true that time study develops these requirements, but these requirements should be attended to first or time study will in a way be wasted.

Accounting. To have reasonably accurate costs is both desirable and necessary, not only as a means of efficiency but also to eliminate unprofitable articles of manufacture and produce the more profitable ones. This system requires methods of factory accounting which interlock with routing system, time keeping, and stores, requiring a close coöperation with the stores, inspection and producing work of management.

Proper distribution of overhead charges, accurate time study and charges, accurate material charges, controlled from a central point in which all is charged against its proper account by the same mental interpretation is the only accurate way, combined with a symbol method, not too complicated, which places the burdens where they belong at the time the expense is incurred.

Stores. Raw and worked materials must be properly accounted for, and the truthful conduct of a stock room is important and necessary. Stores records should be complete and show enough information to gage efficiency without being so complicated and cumbersome as to require more time to tell the story than an inventory.

Maximums and minimums should be so proportioned as to allow a minimum amount of capital being tied up in fixtures.

[1] Trans. Am. Soc. M. E. vol. 24, p. 1337

stock, raw materials and work in process as well as all materials spoiled or defective. A graveyard in a store in plain sight of all is a mighty object lesson to the management and workmen.

The broad or commercial interpretation of efficiency engineering, or scientific management, is profitable management, in which the final measure of success is the return on the investment. As this return is dependent on other than labor elements, it would seem but fair that we carefully and earnestly consider the fact

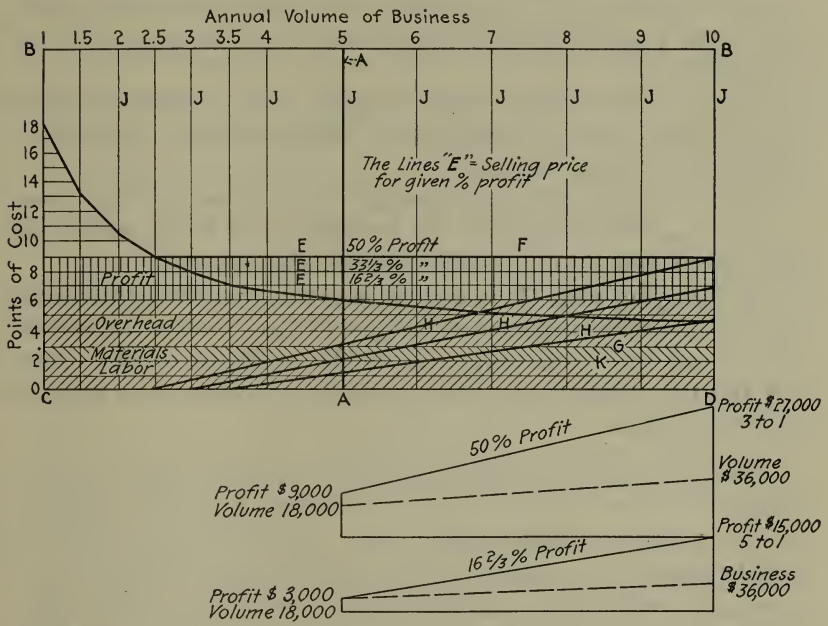


FIG. 17 DIAGRAM FOR DETERMINING COST OF LABOR, MATERIALS, OVERHEAD AND PROFIT

that no man can do efficient work at a machine unless the conditions are made right for the performance of such work and that each function of the industrial organization requires as close, if not closer, attention on the part of the management as do the workman and the machine.

Explanation of Diagram. Variations in expense and cost of labor and materials are practically negligible, but each factory or unit of output should have a separate diagram based on actual conditions of costs of flat labor, materials, overhead and profit.

In Fig. 17 *BC* shows the points of cost; *BD* the points of profit; the different percentages of profit are shown by *EE*.

To use the diagram, assume first that the product is sold at a certain price. Then select *E* to correspond to the profit for these price points. Following this line to the left until it intersects the curve, will give the point where the year's business will break even. To illustrate with 50 per cent profit, it will be necessary to do a business of 2.5 per year in order to break even. On an annual business of 5, the profit would be 3. With an annual business of 10, the profit would be 9. Then the dollars and cents profits would be those made on 5 multiplied by 9 and divided by 3.

The line *HH* is drawn from the point of no profit to the point of maximum profit on the basis of 10 for the year's business, or double the average business.

For any given volume of business of the line *BB*, the points profit could be obtained by the intersection of the lines *JJ* with the line *H* and read out to the scale on the line *BD*.

By studying Fig. 17, it will be seen that the cost of materials and the overhead expenses are so great in proportion to the labor performed that more emphasis should be placed in this report on the management end of the factory than on the workman and the machine.

Fred. W. Taylor. The preparation of this report has evidently involved careful research, followed by a close analysis of the materials gathered in. The viewpoint from which the whole subject is examined is new and original. Most writers upon this subject have emphasized the necessity of reducing to a science the knowledge which in the past has been in the heads of the workmen. The change from rule of thumb to scientific knowledge has been largely dwelt upon, and its importance pointed out. The thought of the committee, however, centers mainly about transfering this knowledge to the workman after it has been acquired by the management; and from this viewpoint scientific management is very properly summarized as " the mental attitude that consciously applies the transference of skill to all of the activities of the industry."

The committee very properly call attention to the significance of the change in the mental attitude of both sides which takes place under scientific management. This would seem to be the most vital element in scientific management. It was,

indeed, looked upon as of such importance that, during the hearings before the House Committee " to investigate the Taylor and other systems of management," man after man came from the shops which are being run under scientific management, to testify that the very essence of this system lies in the great mental change which comes both to the management and to the workmen. In fact, with but few exceptions, these men testified that without this complete mental revolution, scientific management could not exist.

This mental change is great and far-reaching. It means essentially a change from suspicious watchfulness and antagonism and frequently open enmity, between the two sides, to that of friendship, hearty good-will and coöperation. It means a change from the old belief that the interests of employer and employe are in many respects necessarily antagonistic, to the firm conviction that the true interests of the two are mutual.

This feature of scientific management is of such importance that it seems desirable to make one of the causes for the change in mental attitude a little more clear. The following illustration may help to do this:

Into the manufacture of any article there enter two items of expense, the cost of the materials of which it is composed, and the cost of what are commonly called " overhead expenses " or general expenses, such as the proper share of power, light, heat, salaries of officers, etc.

Now, if these two items of expense, cost of materials and general expense, be added together and their sum subtracted from the selling price of the article, we have what is called the " surplus." And it is over the division of this surplus between the company and the men that most of the labor troubles and disputes have come in the past. The men want as large a part of this surplus as possible in the form of wages, and the company as large a share as possible in the form of profits. And in the division of the surplus, under the older systems of management, both sides have come in many cases to look upon their interests as truly antagonistic.

A part of the great mental revolution that occurs under scientific management is the complete change in viewpoint of both sides as to this surplus; from looking upon the *division* of the surplus as the important question, they both come to realize that if, instead of pulling apart and quarreling over it, they join

together and both push hard in the same direction, they can make this surplus so large that there is no need to quarrel over its division, because each side can get a far larger sum than they had ever hoped to get in the past. And each side realizes that this result would have been entirely impossible without the hearty coöperation of the other. The workmen see clearly that without the constant help and guidance of those on the management side, they could not possibly earn their extra high wages, and the management see that without the true friendship of the workmen their efforts would be futile, and they are glad to have their workmen earn much higher wages than they can get elsewhere.

The introduction of scientific management is, so far as I know, the first large movement in industrial history in which a great increase in the output of the workmen has been at once accompanied by a large increase in the earnings of the workmen. The great increase in output resulting from the introduction of labor-saving machinery was not at once accompanied by much if any increase in the wages of operatives, and in many instances the introduction of labor-saving machinery resulted in paying lower wages to the operatives of these machines than had been received by similar hand workers before the introduction of machinery. The main profit was absorbed at first in almost all cases by the manufacturers, who introduced the new labor-saving machinery. In the end, of course, the operatives, along with all the rest of the world, have profited immensely through the introduction of labor-saving machinery.

One of the most notable features of scientific management, however, is that the group of men who have introduced it have insisted that the workmen coming under its principles should at once be paid from 30 per cent to 100 per cent higher wages than they could get elsewhere. This fact is not appreciated by the general public, and largely because the labor leaders, consistent in their fight against the introduction of any labor-saving device, have seen fit to misrepresent far and wide almost all of the good features of this system; and in doing this they have strenuously denied that the workmen coming under scientific management are paid higher wages than heretofore. Quotations such as the following, taken from the famous circular distributed by President O'Connell throughout the Machinists' Union, are typical of this misrepresentation by labor leaders:

Wherever this system has been tried it has resulted either in labor trouble and failure to install the system, or it has destroyed the labor organization and reduced the men to virtual slavery and *low wages*, and has engendered such an air of suspicion among the men that each man regards every other man as a possible traitor and spy.

The installation of the Taylor system throughout the country means one of two things—i. e., either the machinists will succeed in destroying the usefulness of this system through resistance, or it will mean the wiping out of our trade and organization with the accompanying low wages, life-destroying hard work, long hours, and intolerable conditions generally.

In answer to this statement of O'Connell's, however, in the sworn testimony given before the House Committee to investigate the Taylor and other Systems of Management there was presented a schedule of the present wages in comparison with the past wages of all of the workmen who had been more than 12 months in the employ of a company which was using scientific management. This statement showed that the workmen were then receiving on an average 73 per cent higher wages than when they first came under scientific management. The list of employes included all kinds, even the colored men who helped move the materials around the floor of the shop, and the sweepers, etc.

The testimony also showed that the company, after paying this 73 per cent increase in wages, found itself better off than it was under the older type of management. That fewer workmen were turning out three times the output formerly obtained and that the selling price of machines manufactured had been reduced 25 per cent.

It is object lessons of this sort which are rapidly convincing those who investigate scientific management that the interests of both sides are mutual, instead of antagonistic.

The historical portion of the report shows careful study, and is evidently the result of much research. In certain particulars, however, it is somewhat misleading; that portion of it, at least, which includes the quotations from Adam Smith, etc., and particularly Tables 1 and 2, given in the Appendix.

Although the fact is not specifically stated, still the general impression from reading this part of the report is that "time study," which is the foundation for "the transference of skill from the management to the men," was practically carried on in 1760 and in 1830, as it is now under scientific management. This is, however, far from the truth, and in the interest of his-

torical accuracy it may be desirable to make a statement as to the beginning of "time study," although I realize that questions as to who started time study, and when it was started, are of very little consequence, the important questions being, what is time study? and, how shall we make it more useful?

Time study was begun in the machine shop of the Midvale Steel Company in 1881, and was used during the next two years sufficiently to prove its success. In 1883, Mr. Emlen Hare Miller was employed to devote his whole time to "time study," and he worked steadily at this job for two years, using blanks similar to that shown in Par. 367 of "Shop Management."[1] He was the first man to make "time study" his profession.

It is true that the form of Tables 1 and 2, given in the Appendix to the Committee's report, is similar to that of the blanks recording time study, but here the resemblance ceases. Each line in Table 2, for instance, gives statistics regarding the average of the entire work of an operative who works day in and day out, in running a machine engaged in the manufacture of pins. This table involves no study whatever of the movements of a man, nor of the time in which his movements *should* have been made. Mere statistics as to the time which a man takes to do a given piece of work do not constitute "time study." "Time study," as its name implies, involves a careful study of the time in which work *ought* to be done. In but very few cases is it the time in which the work actually was done.

Previous to the development of "time study" in the Midvale Steel Works, there have in all probability been many instances in which men have carefully studied and analyzed the movements of other men, and have timed them with watches. (No such instances have, however, come to my personal attention.) Any such former work was without doubt confined to isolated cases, and was of short duration; and (most important from the historical point of view) it did not lead to the development of a new trade, or, more properly, to a new scientific occupation, "the profession of time study."

Any former efforts of this kind would bear the same general relation to the time study done in the Midvale Steel Works that the many early attempts at flying bear to the work of the Wright brothers.

The Wright brothers started "man flying."

[1] F. W. Taylor, Trans. Am. Soc. M. E., vol. 24, p. 1436.

The Midvale Steel Works started the "profession of time study."

(I do not of course intimate that the two developments are of equal importance.)

Time study is the one element in scientific management beyond all others making possible the "transfer of skill from management to men." The nature of time study, however, is but imperfectly understood and it is therefore important to define it clearly. "Time study" consists of two broad divisions, first, analytical work, and second, constructive work.

The analytical work of time study is as follows:

a Divide the work of a man performing any job into simple elementary movements.

b Pick out all useless movements and discard them.

c Study, one after another, just how each of several skilled workmen makes each elementary movement, and with the aid of a stop watch select the quickest and best method of making each elementary movement known in the trade.

d Describe, record and index each elementary movement, with its proper time, so that it can be quickly found.[1]

e Study and record the percentage which must be added to the actual working time of a good workman to cover unavoidable delays, interruptions, and minor accidents, etc.

f Study and record the percentage which must be added to cover the newness of a good workman to a job, the first few times that he does it. (This percentage is quite large on jobs made up of a large number of different elements composing a long sequence infrequently repeated. This factor grows smaller, however, as the work consists of a smaller number of different elements in a sequence that is more frequently repeated.)

g Study and record the percentage of time that must be allowed for rest, and the intervals at which the rest must be taken, in order to offset physical fatigue.

The constructive work of time study is as follows:

[1] Recording these movements so that they can be readily found is the most difficult element of time study. The writer threw away his first two years of time study because it was so poorly indexed that he was unable to find the elements when he needed them.

h Add together into various groups such combinations of elementary movements as are frequently used in the same sequence in the trade, and record and index these groups so that they can be readily found.

i From these several records, it is comparatively easy to select the proper series of motions which should be used by a workman in making any particular article, and by summing the times of these movements, and adding proper percentage allowances, to find the proper time for doing almost any class of work.

k The analysis of a piece of work into its elements almost always reveals the fact that many of the conditions surrounding and accompanying the work are defective; for instance, that improper tools are used, that the machines used in connection with it need perfecting, that the sanitary conditions are bad, etc. And knowledge so obtained leads frequently to constructive work of a high order, to the standardization of tools and conditions, to the invention of superior methods and machines.

It is unusual to make a study such as this of the elementary movements of the workmen in a trade. The instances in which this has been done are still rare[1]: And it would seem that this must be due to a lack of appreciation of the great power which is given to the man who possesses a knowledge of the time value of these elements, and also to a lack of appreciation of the large variety of work to which these elements apply. How many men, for instance, know that a man who has received his education in "time study" through analyzing their elements of the movements of machinists engaged in manufacturing conveying and hoisting machinery, can go with this knowledge into another establishment manufacturing machinery not in the most remote degree resembling hoisting machinery, and there use this knowledge to fix accurate daily tasks for the machinists? Yet during the past year and more, a young man trained in time study in the Link-Belt Works in Philadelphia, has been setting the daily tasks in one of our arsenals manufacturing a large variety of

[1] Most of the men who have made what they call "time study" have been contented with getting the gross time of a whole cycle of operations necessary to do a particular piece of work, and at best they have thrown out the time when the workman was idle, or evidently purposely going slowly.

war materials, including the great disappearing gun carriages used in our coast defence.

Surely, when the significance of such a fact as this is appreciated, companies employing machinists, even though they may be manufacturing radically different kinds of work, will join together in studying the rudimentary elements of the machinists' trade and then in publishing this knowledge so that it may be available for hundreds of companies, where now it is the private property of the few concerns who have had the patience and the courage to be pioneers in this field. It is the lack of published data regarding the time required to perform each one of the elementary operations in our various trades (more than any other element) that makes the introduction of scientific management such a slow process.

If we accept the committee's definition of the new management as the " transference of skill from the management to the men," it is evident that the management cannot transfer knowledge and skill until they themselves possess this knowledge, and up to this time each new company introducing scientific management has been obliged to obtain this rudimentary knowledge through their own analysis and study, a very slow and tedious process.

Seventeen years ago, I predicted, in a paper read before this Society,[1] that books would be published similar to our engineering handbooks, embodying a time study of all of the elementary operations occurring in our various trades; and, was then greatly laughed and sneered at for making this statement. Only one such book has as yet appeared, but I wish to repeat my prophecy with more emphasis even than before—that hundreds of books of this sort will be published in the future, and in the not far distant future. These books will make possible " the transference of knowledge and skill from the management to the workman " on a large scale throughout the country, and the introduction of scientific management will then indeed proceed at a rapid pace.

F. G. Coburn[1] said that the committee have expressed in their report the " human interest " side of scientific or labor-saving management, which some of its critics say it lacks. The human part of industrial management, it seemed to him, was

[1] A Piece-Rate System, Trans. Am. Soc. M. E., vol. 17, p. 856.

the use of certain psychological or psychic influences to cause employes to work hard, loyally and intelligently, and the degree of success attained depended largely on the number of these influences that were correctly used. They have never been clearly defined and are not clearly understood, even by those who use them. When they are defined, the industrial manager will have a set of proximate rules for a game which is at present played without rules.

The manager who supports his unconscious use of these psychological influences with known methods of labor-saving management, i.e., worked-out details based on the basic and regulative principles stated in the report, applied to his own plant, is bound to be better off than if he did not.

The system of management is but the tool of the manager. Alone it is useless, may even be dangerous; the management without it is handicapped, whereas the system, backed by an active mind continuously using it, is the effective combination. And labor-saving management is a system that forces the executives to manage.

As at present understood, the expressed principles underlying and regulating labor-saving management do not completely solve industrial management. They may only postpone the strike problem. A prominent investigator of business and industrial problems expressed this by saying that there is no assurance that the increased reward offered to the worker for attaining high production is the worker's correct share of the increased profit. He has his own misgiving as to what would happen were the greater part of a given labor market employed under labor-saving management.

It would seem that coöperation between employer and employes, so necessary to the successful practice of labor-saving management, is based on a bargain subject to change; and that successful readjustment of the bargain would be required from time to time, to insure the continuance of such coöperation. This may settle itself. A noteworthy example of readjustment initiated by the employer in favor of the employes has been recently brought to his attention. He expressed the opinion that the committee's report opened the way to study of psychological principles—or psychic, perhaps—that are involved; and the hope that some day may see a committee of this Society appointed to take up this investigation.

Carl G. Barth. I am gratified with the report as an endorsement of the kind of work I am permanently engaged in; particularly as I find among the names of the members of the committee who have signed the majority report, that of a personal friend who, only through this investigation has become a convert to scientific management. I am unable to take as broad a view of the matter as the committee have attempted, as I am tied down too closely to the daily details and difficulties of the practical introduction of scientific management to devote much time to its broad historical and economic aspects. Part of the report, also, has a rather amusing side for those of us who now for several years past have been working with some success in this field, in that it is virtually a declaration to the effect that the committee have assured themselves that there really is such a thing as "scientific management," and that it does accomplish some of the things, at least, which its exponents allege that it does. It reminds me of the farmer who came to town in the early days of the automobile to assure himself that there really was such a thing as a horseless wagon not requiring tracks to run on, and that it actually did carry people around in the streets without running wild and upsetting everything in its way, except when in the hands of a driver possessing more ambition than experience and sense of responsibility.

As regards the minority report, I see no good reason for its coming into being, as it contains nothing that I cannot agree to, and to my mind nothing that it recommends conflicts with the majority report.

It merely asserts that there are places in which, and conditions under which, the introduction of scientific management would not be a paying proposition; a matter that I do not believe anybody will dispute. In fact, experience has long ago taught me that there is hardly an establishment in which there is not some department or corner of the works which is not better left almost untouched by the new order of things, except in the matter of having its relations to the rest of the works properly dovetailed into this.

To take the automobile for a second illustration, I have never heard of anybody recommending its use to the extent of declaring that walking is no longer a profitable and sensible method of locomotion, to be eventually given up entirely; though we also know that some automobile enthusiasts do use the automo-

bile at times and in places where walking would be more appropriate.

As regards the attempt of the committee, and others, to give a more appropriate name to what we all now pretty well understand by scientific management, I like to state that I have until recently preferred to refer to what I am trying to do, as the "Taylor system of management;" and that Mr. Taylor himself was the first one to discourage me in thus using his personal name in that connection, and to suggest substituting therefor the term scientific management. However, as I am not only not ashamed of, but on the other hand exceedingly proud of, being accused of being Mr. Taylor's most orthodox disciple, I have stuck to the former until recently; though since numerous imitators have invented a number of substitute names, so far as I can see to no practical purpose, I have become more favorably inclined to the use of the term scientific management. Accordingly, I now refer to myself as an exponent of the "Taylor system of scientific management," and feel that in doing so I have just about the whole thing.

In his discussion Mr. Gillette tries to point out that Mr. Taylor and his disciples have not covered the whole field and mentions as an example, cost and accounting as a matter to which they have paid no attention. However, the mere fact that Mr. Taylor has not written on every subject of his activities, must not be taken as a proof that he has neglected any important matter connected with the management of an industrial institution. The fact is that Mr. Taylor's disciples also owe him a great debt for the cost and accounting system he has handed over to them, one which for completeness and flexibility stands absolutely unequalled even today, in spite of the great attention this subject has recently received.

Mr. Church also mentions that the study of machines is a matter thus neglected. However, this is also a subject which Mr. Taylor gave serious attention over 25 years ago, or about the same time that he first took up the study of the art of cutting metals, and this has played one of the most important roles in the subsequent work done by myself in the perfection of some of Mr. Taylor's methods.

Again, one speaker suggested that there should be a special purchasing agent for buying labor, as if that too were a brand new idea, whereas Mr. Taylor has also pointed out the desirabil-

ity of that, and as long ago as 1895 gave us an example of it in a man employed by some Western company at a salary of $5,000 a year, who had the buying of labor down to a fine art.

Mr. Gantt has told us about the disastrous results of changing to piecework from his bonus system of paying, in a plant systematized by himself, and I fear that what he said might have been misleading to some of the audience.

There is no question that piecework based on careful time study is the best and most just form of contract between employer and employe, and though I have not introduced Mr. Taylor's differential piece rates for some years past, I still believe that system is the best suggested to date. However, it is so difficult to get employers to wait until all such standards have been provided, and such time-study men have been trained, as are absolutely essential for the introduction of a piece-rate system, that we are usually forced by circumstances to beg the question and to resort to the expedient of a bonus or premium system.

These systems have the advantage of any piece-rate system, and particularly the differential, in that they may be established and give some satisfaction to both parties concerned, long before it would be possible to make a success of a piece-rate system.

However, there is no doubt in my mind, that the differential piece-rate system conceived and used by Mr. Taylor some 30 years ago, is the ideal of all the schemes suggested to date for paying workers in any other way than by straight day wages.

Regarding the use of a moving picture machine for motion- and time-studies, it looks to me to be practically the same thing as was offered to the Watertown Arsenal at the time we took up that part of the work there. I do not remember the inventor's name, only that he was connected with the Navy or with one of the navy yards in one capacity or another. While the scheme was interesting, it was declined after a conference between the commanding officer Mr. Merrick, and myself, particularly on the opinion expressed by Mr. Merrick, who is one of the most experienced time-study men in the country, and because of the anticipated great expense both of procuring and operating such a machine. Perhaps this was a mistake, for no doubt the machine can be successfully used for motion study, and hence for the elimination of useless motions, but I am still open to conviction regarding the use of any machinery of this kind in connection with time study. The two main difficulties in time

study are: first, to judge when a worker is working at a proper rate and to make proper allowances when he is not; second, to make proper allowances for necessary rests, etc.; and I do not see how any kind of machinery can help us in these difficulties.

Alex. C. Humphreys. We should discuss frankly the questions that come before us. Professor Thompson in his remarks made this statement: "Referring to the problem of wages, the solution of this problem, if there is one, is not to be looked for from engineers. When an engineer wanders into the field of economics he is apt to bring about the same diverting spectacle that the economist would who would undertake to expound the principles of machine design."

I think that this statement should be met in a meeting of this kind, not in the interest of the engineer so much as in the interest of truth; and certainly so in this particular meeting which is called to consider the subject of efficiency in management. This day's work alone, I think, should be sufficient to contradict effectively the statement to which I object. Again, the record of our industrial life, largely directed and controlled by engineers, serves to contradict the statement. A book economist makes his principal error, and, in my opinion, he has many to answer for in this country especially today, in neglecting to consider facts developed in the field of work and so in failing to look on all sides of the question, as necessarily, in my opinion, any man must look; particularly one who is primarily a bookman. Unquestionably, the book economist fails in this regard because he is not qualified by practical experience to arrive at exact conclusions.

Finally, after giving this matter very serious consideration for a number of years and having had, I believe, unusual opportunities to judge through my work as an engineer, an industrial manager, and the president of an engineering college, I do not hesitate to say that some of the most inefficient guides as to questions of economics are certain of the professors of economics in our colleges and universities. I trust that Professor Thompson will believe that in this nothing personal is intended.

David Van Alstyne[1] in a communication endorsed the minority report of Mr. Vaughan. Referring to Par. 1*a*, he said that while it may be possible that the application of the newly dis-

[1] 105 West 40th St., New York.

covered principles of management to railroads would result in a million dollars a day economy such a statement has no practical value because the application of these principles is impossible at present and will be for many years. The investment required to make possible such economy would be enormous and the change of attitude on the part of railroad owners, operators and employes before such investments and changes in methods are possible will be long deferred.

Referring to Par. 1*c*, any system of management which cannot be made acceptable to organized labor is doomed to failure. There is probably no greater problem before scientific managers than to convince labor unions that scientific management has advantages for them as well as for employers.

The term scientific management seems inappropriate because it tends to emphasize the contrast between a system presumed to be based entirely on scientific principles and one in which presumably nothing is done scientifically. Neither wholly scientific nor wholly unscientific management exists and the degree to which management is scientific is the extent to which it endeavors to know what its conditions are, what they should be and how to bring the actual to the standard.

Hugo Diemer. The report of the committee indicates that in their opinion there is no system of management any more than there is a system of music, chemistry or education.

The committee are to be commended for a strong effort put forth to get at the fundamentals. It should be borne in mind, however, that it is possible to go to extremes in our search for basic principles. For instance, we might consider that the basic principle of mathematics is measurement. If we contented ourselves with this bald statement we would not explain the scope and objects of the science of mathematics as fully as if we extended our definition somewhat.

I believe we should not content ourselves with saying that the principle of division of labor is the basis of manufacturing. If we add to this statement a further one that the principle of division of labor gives rise to such matters as organization, control, specialization and functionalization, we are adding to our definition the sort of matter which we would consider it necessary to add to our definition of the basic principle of mathematics.

Attention is called by the committee to the fact that early writers foreshadowed the planning department, also that motion and time study are not even 19th century products, but that they were known

and practised at the opening of the manufacturing era in the eighteenth century.

The relationship of these isolated cases to the modern science of management is not much more definite than Jules Verne's prediction of the submarine and the aeroplane to the modern perfection and utilization of these ideas. Even in King Solomon's time we had pretty well regulated subdivision of labor, also the eight-hour day with Saturday half holiday and pay checks.

There is no present day science that was not foreshadowed by disconnected parts in earlier centuries. For this reason I would change the definition which the committee have given for the second basic principle, which they designate as the transference of skill. The committee illustrate as examples of this principle the transference of the skill of the inventor to the power-driven mechanism, also the transference of the skill of the real expert in management who trains and directs the persons who are permanently to manage. I believe this element can be better defined by stating that it is the conscious application of scientific methods to all phases of industry, or the conscious effort to build up a science of industrial management. The real science of management will apply research methods, not only to designing and testing but to all phases of industry.

Henry P. Kendall[1] outlined the experiments which have been worked out by the company with which he is connected, in the purchasing of labor, and spoke of the work of the employment man as a particular function in scientific management.

In this establishment this man engages the employes and tries to apply to them whatever physiological test he can to determine the kind of work each is best fitted for, or whether any particular one is fitted for the position for which he applies.

In employing women and girls, all are hired subject to the approval of the factory nurse, and if there is any question as to the health of men applicants, they are likewise passed upon.

So far as any person can judge, employes are given the kind of work for which they are best fitted.

The employment man's responsibility does not cease when he has hired the man. He has to follow his record in his work, and adopting a suggestion that came from James M. Dodge, cards are used to preserve the record of each employe. Four times a year information on each employe is added to these cards, as to how he is performing his work, his aptitude for it,

[1] Plimpton Press, Norwood, Mass.

his skill and earning power, and general deportment. He keeps in touch with each foreman or gang boss, and gets a signed report on each employe for these card entries.

The hiring and discharging and matters of discipline of the factory are all concentrated in the employment man, which prevents the injustice that comes from a gang boss or foreman discharging a man in a moment of anger and saying that he is no good. Should he say that a man was no good, and recommend his discharge, a written report from several quarters might contradict the statement, if it were unjustly made.

The training of young workers is especially important. There is nothing so discouraging to a boy as to be set to work at the end of school by an average foreman and forgotten. He may be in the wrong place and not fitted for that kind of work. He has, in the employment man, one to whom he can go for advice in regard to his present employment, for suggestions in regard to outside education or instruction, one who is in sympathy with him, and with whom he can rest his grievances and feel secure that he will not get in wrong with the boss.

The foremen at first thought they would lose their authority when they could no longer hire and discharge, but now they feel that it rather helps their authority to have the employment man to send to in cases of discipline. When a man is discharged by a foreman, although perhaps for cause, this may be done in an angry manner, and he may leave defiant, and with a determination to "get even." If instead, he is called to the office of the employment man and an explanation is given him as to why his services have not been satisfactory, and he is made to see that he is at fault, and if this is done in a quiet frank way, he leaves with an entirely different feeling. Under this present scheme, the employe is given an opportunity to register just complaints against a foreman, if he has any.

The purchase of machines and equipment has been strongly emphasized. The purchase of labor is more important, but is given less attention. Then, too, the main functions of scientific management can be used only in combination with other functions. This function can be adopted in any concern with equal advantage.

Tracy Lyon suggested that if there is any lack in the report it is in the consideration of the reasons for the vigorous opposi-

tion to the new element of management which is mentioned and the absence of permanent results from some conspicuous efforts in this direction. The subject of so-called scientific management is often clothed in glittering generalities which offer no practical suggestions to the average man, but it really means no more than that the manufacturer, operator or business man shall take the time or employ others to consider analytically every condition of his business and every step in his operations rather than to push ahead by mere force of weight. Looked at in this light it would seem that any real opposition to the "new element" must be to the form rather than to the substance and to ill-advised and hasty efforts of "systematizers" in name only. We have yet to discover any royal road leading to efficient management other than through painstaking and persistent effort.

It is difficult to understand a charge, that if a manufacturer's tools are put in the best operative condition and so arranged as to minimize the movement of material, if provision is made for material in definite and sufficient quantities, small tools and other necessities are on hand before the commencement of a piece of work, and a liberal and intelligent measure established for the payment of the workman, the result will not be to the benefit of the employer, the employe and the consumer. The function of a planning system is no more than to provide in advance that the right thing shall be in the right place at the right time, and to distribute the work to the proper tools in such lots that twelve men will not be nursing their jobs with a small amount of material ahead of them, when there is not more than six can take care of. It does not seem as though such a provision could do otherwise than reduce the cost to the consumer.

Any plan of wage payment can be used whether bonus, piece or day work without interfering with the principles of scientific management, provided there is a clear understanding that the employers are willing to pay liberally for a fair day's work. Some of the systems may have been handicapped by the effort to convey instructions to the workmen in too elaborate a manner. The prime requisite is that time study men and instructors should be thoroughly practical and experienced.

Dexter S. Kimball. Ever since the beginning of the present industrial era there has been a growing tendency to separate mental and manual processes and to again subdivide these by

the application of the division of labor. The engineering and drafting department was the first direct result and is the best existing example of the separation of the mental from the manual. While the art of industrial management has lagged behind scientific design for natural reasons, its fundamentals are well known and basic data will no doubt soon be available so that we may expect to see it on a more scientific basis.

It is to be noted that the process of "functionalizing" any branch of industry involves no new principle. It is an extension of the general tendency to separate mental and manual processes and motion and time study are not new discoveries as the report clearly points out; but this should detract in no way from the credit due those who have so ably called our attention to the possibility of their more extended use. The ultimate extent to which these principles will be carried and the rate of progress that they will make will depend, however, not on their inherent value or effectiveness alone, but on the growth of public opinion that permits or demands them.

One phase of the discussion touched on in Pars. 62 and 63 deserves more than passing thought. Changed industrial conditions resulting from the separation of the worker from the ownership of his tools have introduced a difficult problem in the matter of the compensation of the employe. The efforts of such men as Halsey and Towne were the first steps toward new methods of rewarding labor. All so-called *systems* of management are based on the extension of the principles already mentioned with the addition of some special philosophy of management which invariably includes some variation of these methods of rewarding labor, mixed perhaps, with certain altruistic ideas on the relations between employer and employe and the general uplift of the workman. These altruistic ideas are fine and helpful, though not new, as they were introduced and used by Robert Owen more than a century ago.

Success in the use of these combined methods, whereby better wages and better conditions have accrued to small communities, has led some over ardent advocates of these methods to the hasty conclusion that they really offered a solution of the industrial difficulties which press so heavily upon us. Statements similar to the following are often seen:

"The increase in wages which accompanies this type of management will largely eliminate the wage question as a source of dispute."

"The one cure, the only one for the condition which confronts us is to increase the efficiency of the producer."

That is to say increased productive power *necessarily* means increased profits to the actual producer. But a diligent search through the literature of efficiency engineering will fail to disclose a single new regulative principle bearing on the equitable distribution of the fruits of industry either between the employer and employe, or between the industry as a whole and the ultimate consumer.

Stripped of all sentiment these new methods are exactly similar in operation to labor saving machinery (Par. 37) and their initial and ultimate effects must be similar. They will enable those who first adopt them to obtain some commercial advantage and to pay better wages than their competitors for a time at least. They will also result, if universally adopted, in a considerable increase of productive power; but they in no way affect the fundamentally inherent advantage which must always accrue to those who control the tools of production. They can in no way affect the principle that knowledge of all kinds tends to become common property and for that reason no labor saving device can forestall competition for any extended time. They can in no way affect the law of supply and demand; and even if all the manufacturers in the country were to adopt these methods at once we should continue to see in many places at the same time, storehouses filled with raw materials, idle factories equipped with the finest tools, and people walking the streets without food or clothes and yet willing to work. Labor-saving machinery opened the door to tremendous *possibilities* for humanity far greater in proportion than can ever be promised by labor-saving management; but it brought with it no self-regulating principle which insured equitable distribution of these advantages; and labor saving management is at best a small extension of a great movement.

It is true that all classes of people have profited by the use of labor-saving machinery, the actual producer as much as any, and it also follows that in the long run we shall all profit to some extent by increased productive power. But the industrial problems with which we are now wrestling will all be with us till we have made some changes in the basic principles on which we now distribute the results of our labors.

True, all the new systems lay stress on the "square deal" and

the coöperation of employer and employe, but these are old as mankind and have always been available as a basis of economic justice in common with all other welfare work. But these measures are optional with employer and employe and do not constitute basic principles. They are exceedingly useful and beneficial as they serve as a means of better understanding of the dual difficulty and as a basis of corrective legislation, which after all is a reflection of the sentiment of the community and grows with it. It is neither fair nor wise, however, to attribute powers and advantages to efficiency engineering which it clearly does not possess. I am not sure but what a small readjustment of some other of our economic problems would do as much for us as increased productive power. We need and need badly some *scientific distribution* and we shall be fortunate indeed if some of the reactive influences now at work on our social and industrial organization will offer some help in this direction before the strain becomes unbearable.

R. R. Keely briefly summed up the economy from the application of scientific management through placing the control of production in a central office. By this means customers are prevented from becoming discontented through having an order handled improperly or overlooked, and there is a saving in time in getting out important orders, in the nervous energy of the management, in material through preventing waste, in losses due to ignorance of the actual condition of the plant in the old system, and in labor due to the planning of all work in advance of its performance, and the best workmen are induced to remain through the creating of contentment and the offering of higher pay.

The "mental attitude" in scientific management, to which the committee refer, is new in that the individual and his workplace are considered, instead of the shop as a whole, as a unit. So far as he knew there had never been a planning room for the handling of the entire work of an industry from a central office previous to the introduction of scientific management by Mr. Taylor.

Joseph A. Bursley called attention to the fact that this was the first time the Society had devoted an entire session to the consideration of scientific management. He was particularly

interested in the possibility of applying the principles of this subject to works of educational institutions, especially technical schools, and felt that there was a great field for applying this work in teaching in the engineering schools. The men in the shops connected with the universities, for instance, should be shown the best way of doing each particular piece of work, so that after they are graduated, they may transfer this knowledge to others.

This same scheme could be followed in the laboratories where various tests are made; and even in the class room in the solution of problems of various kinds there is always one way which is the best, requires the least time, and gives the smallest opportunity for mistake.

William Kent. I hesitate to criticize so splendid a report, but I do object to the statement in Par. 37, "The expression 'labor-saving management' better conveys the meaning of the movement." Labor-saving management is but a fraction of the new movement. As I conceive it, the whole question is industrial management, and there are three kinds of industrial management, which some one has called, traditional or unsystematized, transitional or systematized, and scientific management. This third type is industrial management carried on in a scientific manner, and it refers not only to labor-saving management, but to saving of fuel, capital, machinery, wear and tear, lubricants, and everything that enters into the product. I think if a scientific expert were called in by the management of a factory and they should say, "We want your services to show us what we ought to do," he would have carte blanche to go into every department as well as the labor-saving department.

In regard to the statement that has been made to the effect that an engineer who dabbles in economics is going outside of his field, I wish to call his attention to the fact that the first paper mentioned in Appendix No. 2 of the paper, No. 207, written in 1886, was on the topic, "The Engineer as an Economist." The engineer was an economist in 1886 and long before that. Part of the education of the engineer, and no small part, is education in economics. The thoroughly educated engineer must be a political economist and when he dabbles in questions of economy, he is right in his own field and is not going outside of it at all.

H. H. VAUGHAN explained his objections to the majority report. He differed from the report because it definitely advocates certain things, such as the bonus system, the planning department, functional organization and time studies, as constituting the new way in which to run industrial establishments. No one could value Mr. Taylor's work more than he, but his methods had their limitations like most new discoveries. It is a mistake to think they will revolutionize everything and supersede all other systems.

One thing in the report was very interesting to him, the remarks of James Nasmyth, quoted in Par. 9, in which it is stated: "The characteristic feature of our modern mechanical improvements is the introduction of self-acting tool machinery. What every mechanical workman has now to do and what every boy can do, is not to work himself, but to superintend the beautiful labor of the machine. The whole class of workmen that depend exclusively on their skill is now done away with." He thought that a good instance of how a man's ideas that a new discovery is going to take the place of everything that comes up is frequently limited by his subsequent experience. There are more skilled men in Nasmyth's shop today than there were at the time that Nasmyth wrote these comments.

In his minority report Mr. Vaughan used the phrase, "art of management." There is essentially an art in these things, and one cannot give the rules that make an art. There is more in the creating of a successful shop than the studying of a set of prescribed methods. There must be the peculiar intuitive judgment that comes from experience and from the contact with men.

He said further: "Transference of skill," referred to in the report, means in one place doing away with skill, in another the improvement of skill, and develops into the idea of telling the men how to do everything. It is commercially absurd to assume that we will reach the point where we instruct the men in everything. I believe in experts in the shop, but the experience of the men in the shop must be taken into account. Anybody who has started to build a machine or produce any other class of work with new and unskilled labor, has found to his cost the value of the skillfulness of their men as an asset.

The report advocated functionalizing. Instead, I believe in putting more responsibility upon the foreman, making him the head of the units. Functional management is very much like departmental management on a railroad as opposed to divisional management. The most successful railroads are managed on the divisional system.

We do not expect a divisional man to know everything, but we work through him with the staff department, and want him to be the boss of his unit as far as possible. In that way we certainly get results, quicker action in the case of emergency than where the departmental system is in use and where one must go through a lot of routine to get results. Suppose we have a rush order. One man is put in charge of the job, who follows it through department after department, to see that the work is properly done. We could not do that if each department had to be called upon through its head to coöperate to obtain the results. The best results in our shop have been obtained by putting the work up to the superintendent or foreman and leaving the matter in his hands.

With our piecework system the object is to advise the foreman of what is being done rather than of what has been done. We have obtained good results by putting on a man whose duty it is to advise the foreman from time to time as to what is being done so that he can take immediate action in case any men are failing to make the time expected, as governed by the records in the operation of the shop.

As to the question of fairness in dealing with labor and cutting prices, much of this is really due to the fact that the workmen of today will not stand cutting prices. A few years ago men were less independent and many shops were operated satisfactorily on straight piecework and with cutting prices. There is no use in pretending to be altruistic in this matter. We have been forced to our present position by experience and it is pleasant to know that we can truthfully say to ourselves that it does not pay to cut prices.

We have been building the same class of locomotives for five years under the ordinary piecework system, and by dealing with the men fairly and encouraging them and getting them interested, the hours have dropped from 100 per cent to 68 per cent.

We believe in the piecework system with a guaranteed day rate, and do not use the bonus system, although in my opinion it is better than piecework where the operations are comparatively simple and the conditions standardized. However, the system must appeal to the men, and if they do not understand it the same degree of success will not be secured.

We are using time-study work to a large extent, and I think time studies are of great value in perfecting a shop. On the other hand we had a shop with a pay roll of $4000 a month get into bad conditions. We sent a couple of men there, with a few good demonstra-

tors, picked it right up, and it began to turn out the output. That is better than if we had been depending on a long-drawn-out system; it is a question of personal influence and quick action. The influence exerted in that case will be felt for a long period to come.

As Mr. Gantt has stated, he introduced time studies in our shops, which we value highly. Our results in the time-study work have cost on the average $600 to $800 a month, since we started, and we are now saving $2000 a month on a pay roll of $35,000, contract work. There has been a satisfactory progression with time study, not a revolution; we would not be without it, but, on the whole, it has not been as important as the piecework system.

There is, moreover, certain work in which day work is preferable to piecework. We would not think of putting steam-shovel repairers on piecework. We would rather have them work day work. We must use day work in certain cases and we should then do all we can to help the men to get the results.

We believe that we should not have any more system in our work than we can get along with nicely. We would rather have too little system than too much in the operation of the shop. We think that the best results are secured not only by studying the conditions, but also by studying the men who are operating the shop.

Thomas R. Woolley[1] considered that the benefits from scientific management had not been all that could be desired for the worker, the manufacturer or the consumer. The worker is dissatisfied because of the strain put upon him to increase his speed and the stop watch in particular causes dissatisfaction. If instead of taking the initiative away from the worker, the rule could be reversed and the initiative of the individual encouraged under the guidance of competent executives, the system would have permanent results.

While the manufacturer has been benefited in some cases, in other cases he has been driven into bankruptcy on account of the large overhead burden introduced to carry out effectively the system, particularly where the business fluctuates greatly from year to year and from season to season. Under such circumstances, it is questionable whether it pays to train workers only to let them go during a dull season or to carry them on the pay roll working inefficiently during that period.

Further, the public had not been benefited by the adoption

[1] Efficiency Engineer, Boston, Mass.

of the system and the writer cited a case where a large order for steel was placed, after securing bids from different firms, one of which employed scientific management and whose bid was higher than those of its competitors.

Dealing with Mr. Taylor's example of training men to carry pig iron, he believed that a fraction of the money spent upon labor saving machinery with an electro-magnet in connection with a crane would give better results at less expense.

In considering whether to install scientific management, a manager must determine whether there is enough of one class of work to make it pay him to train men; whether his plant runs continuously throughout the year; whether it will pay to add a corps of men to instruct his workmen; and whether he should consider that it would pay better instead to install the latest improved machinery for the particular class of work to be produced.

Mr. Taylor and his associates are to be commended for stimulating thought along the lines of eliminating movements, etc. Some of these principles can be applied in a plant but all can not be applied in all plants because of varying conditions.

H. K. Hathaway wrote that the essence of this report lay in the phrase (Par. 29), "The mental attitude that consciously applies the transference of skill to all the activities of industry," further qualified by the statement, "the novelty of the new management lies in this transference of skill from the management to the workman.

This viewpoint is so novel that one is led to question whether the older descriptions of the essence of scientific management may not have been wrong. The committee, however, are merely laying emphasis upon one of the elements which has heretofore been looked upon as rather of secondary importance.

Before skill can be transferred, the management must gather in and record the knowledge and skill which were formerly widely diffused in the possession of a great number of workmen, and much of which was an almost unconscious inheritance. Gathering in and systematizing this knowledge ready for use, constitutes the development of a science to replace the old rule-of-thumb knowledge: this is what has been called by other writers the first of the four principles of scientific management, and this point the committee recognize in Par. 47.

The writer's experience leads him to feel that the committee

have not given sufficient prominence to the element of acquisition of skill or "the development of a science" as it has been termed. Such development goes farther than the acquisition of existing skill and results in additions thereto as well as improvement in methods and machinery, and the establishment and maintenance of standards.

It is impossible to transfer skill from the management to the workman without first choosing carefully the workman who is fit to do a particular kind of work, and then training him until he acquires skill: the second of the principles of scientific management. It is not only a transference of skill, but in one sense the creation of skill as well; perhaps a better expression would be the development of latent skill.

The third principle of scientific management has been called "bringing the science and the scientifically selected and trained workman together," which is merely another name for the committee's statement, "the mental attitude that consciously applies the transference of skill to all the activities of industry."

The fourth principle of scientific management has been defined as "an almost equal division of the work and responsibility between the management and the workmen. The management takes over all work for which they are better fitted than the workman, while in the past almost all of the work and the greater part of the responsibility were thrown upon the men." This principle also follows directly the moment the management accept as their duty the first, second and third principles of scientific management. The burden of developing a science where only rule-of-thumb knowledge existed in the past, and the burden of training, teaching and transfering skill from the management to each workman in the place, of necessity calls for an almost equal division of the work between the two sides. So that in the one definition of the new science, by the committee, are implied all of the four principles of scientific management.

It might be proper to point out, in order to avoid misunderstanding, that specialization of workmen on one comparatively simple operation is not necessarily in accord with the best and most economic practice in management. It is a serious mistake and a disadvantage to have operators "unskilled except in a single readily mastered operation" as in the instance cited in Par. 8 of the committee's report. The writer's experience has been that it is not only generally desirable, but in many cases

necessary to economy in manufacture, that each operative become skilled in at least two and generally more of the operations making up a process of manufacture. This is an objection to over-specialization purely from the standpoint of production.

Sanford E. Thompson. Criticism is frequently made of the amount of mechanism required to handle the work of modern management, the number of clerks needed, and the consequent increase in overhead charges, and to a visitor this increase in office is indeed more noticeable than the decrease in number of workmen or the increase in production.

If it were possible in a given establishment simply to make a few time studies and set scientifically accurate rates or tasks so as to obtain maximum production, no objection would be raised to the installation of scientific management, although it is obvious that if such a plan could be arranged, it would have long ago been adopted without resort to laborious study. Yet the criticism is heard so frequently from men of intelligence that it is worth careful consideration.

If we refer to the list of papers selected from the Transactions, and given in the Appendix to the report, it will be seen that those presented earliest discussed methods of setting rates for paying men. Following a short general paper by Mr. Towne in 1886, there are two papers on Profit Sharing, the least intricate form of incentive to the workmen; then from 1890 to 1903, the question of payment is discussed more in detail, the premium plan, piece-rate and bonus systems; then come subjects relating more specifically to the more general problem, the management of the plant. It is evident from these papers that this sequence is not chance, but represents a development, not of theory but of absolute necessity in shop management.

A man who has made studies of workmen and introduced piece rates or tasks on accurate detail time study instead of by the usual "fix and cut" methods, finds that the very requirements of accuracy lead necessarily to standardization of methods. It is necessary to plan out the work of each man, to instruct him in the method so that he may use the least material and accomplish his work in the smallest practicable time, and to install a system of cards and records specially adapted for keeping track of the materials and the time of the men.

The development of what Mr. Taylor terms the mechanism

of scientific management is thus seen. This development indicates that the broad general principles of planning, routing, instruction, scientific analysis, time study and standardization do not represent a mere system for which some other system may be substituted with equally good results, but a plan which is of broad application in management. These things appear to be fundamentally necessary for any organization which proposes to set rates and tasks which will give the men payments based strictly on the actual work which they accomplish, and which will enable them to do the work with the minimum waste of material.

The practical necessity for the introduction of these individual features is very plain to anyone who makes a scientific study of any operation for the purpose of setting tasks.

These statements may be illustrated by one or two examples from actual practice:

Take the trimming of large sheets of paper to make ready for the press. The machine consists simply of a bed plate about 5 ft. square at a convenient level above the floor and an adjustable knife running across this bed plate and operated by power. The operator takes the paper, places it on the bed plate, makes the cuts and transfers the paper back to a truck: a very simple operation, lifting paper, cutting, lifting off.

When the work is studied with a view to putting it on the task basis, the following facts are discovered: The cutter is capable of going through a thickness of about 5½ in. of paper. The size of the paper on the particular machine referred to is large, running up to 40 in. by 60 in. On account of the weight, therefore, the operator has to put it on in several lifts. Since the lift off is easier he is able to take it off in a fewer number of lifts. The number of lifts in both cases is not directly proportional to the weight. It is dependent also on the size, the stiffness, the thickness and the surface finish of the individual sheets of paper. It is readily to be seen, if a man takes six lifts where he should take four, as he naturally will do if he wishes to shirk, the time of lifting and therefore the cost of this part of the work will be 50 per cent too great. It is necessary, therefore, before establishing a task to standardize the plan of furnishing the paper. The paper was always counted, but instead of the usual plan of counting by hundreds, it is now

arranged to divide the piles by slips of paper into individual lifts which a man should take.

For determination of the amount of lift and the other variables, special studies were required. These studies took rather more time and skill than the actual setting of the tasks when this preliminary work was accomplished. After the tasks were finally set and put into operation, it was necessary to throw out of commission the other cutter, which had been working on part time, and the only difficulty experienced was in finding work enough for this one cutter to do. As a result, the cost of the work was reduced 30 per cent with, at the same time, an increase of 40 per cent in the pay of the workman.

This is by no means an exceptional case. It is always necessary first to standardize the machine and the methods. It is this as much as anything else that differentiates scientific from rule-of-thumb methods. Usually the direct results of standardization produce even a greater saving than the actual task and bonus.

Another illustration may be taken from a shoe shop. In the cutting of leather, or in cutting linings from cloth, it is a simple matter to make time studies and determine the time required by the men to pick up the die and make the individual cuts. Piece rates may be easily set, based on the output of each cutter. Suppose, then, that this is done and such rates are set. The men immediately begin to "speed up," but in doing this without any restraint they naturally use more material than is necessary. Now, in the case that the writer has in mind, the raw material used in a certain shop during a week averaged $50,000 in value. The labor cost of cutting this material averaged $280. A 10 per cent reduction in the labor cost amounted to $28, while a 1 per cent reduction in material cost amounted to $50. Instead of gaining money by the setting of rates, the plant would actually lose money, because the men would waste more material. It was necessary, therefore, at the start, before setting any rates to attack the quantity side of the problem, investigate the best plan for layout of dies on the machine and establish some method for fixing the amount of material used by each man. This involved eventually a definite planning of each man's work, so as to show him how to use the minimum amount of material; a routing system, as an absolute requirement, which would properly deliver the materials to him and take away his product; and

a system of instruction and time cards, so as to handle the work in the smoothest manner possible.

It may be claimed that these are special instances which do not often occur, but experience shows them to be very simple cases. Wherever a machine or an operation on construction work is studied thoroughly in order to eliminate unnecessary operations and the waste of materials so far as possible, similar methods must be gone through and a more or less complicated system adopted for the proper handling of the work. Only by such thorough study and investigation can the operation be reduced to a routine which will be essentially automatic.

Frank B. Gilbreth. The report is especially valuable for the reason that it emphasizes the fact, which has long been realized by those engaged in the work of installing scientific management, that transference of skill is one of the most important features.[1] They do not, however, make sufficiently plain that such skill and the experience which precedes it must be measured and recorded before it can be most efficiently transferred.

A better name for scientific management is "measured functional management." It is not sufficient to call it "labor saving management" for it deals with more than labor and labor saving. It is a way for obtaining methods of least waste. It not only saves useless labor, but it improves labor conditions; improves quality of product; prolongs the period of the worker's productivity; conserves, teaches and transfers skill and experience. The committee have caused the Society and the world to recognize at last the importance of the feature of the transference of skill, but they apparently still lack appreciation of the even greater feature of the recording and transference of experience of Mr. Taylor's measured functional management and of micromotion study. Mr. Taylor's system is best described in his writings entitled A Piece Rate System, Shop Management, and On the Art of Cutting Metals, published by the Society, and Principles of Scientific Management, published by Harper & Brothers.

As brought out in the report, the importance of transference of skill was realized many years ago. Studies in division of work and in elapsed time of doing work were made by Adam Smith, Charles Babbage, M. Coulcomb and others, but accurate

[1] See Primer of Scientific Management, F. B. Gilbreth, p. 56; Psychology of Management, L. M. Gilbreth, chap. 8; Motion Study, F. B. Gilbreth, p. 36.

measurement in management became possible when Mr. Taylor devised his method of observing and recording elementary unit net times for performance with measured allowance for fatigue.

It is now possible to capture, record and transfer not only skill and experience of the best worker, but also the most desirable elements in the methods of all workers. To do this, scientific management carefully proceeds to isolate, analyze, measure, synthesize and standardize least wasteful elementary units of methods. This it does by motion study, time study and micro-motion study which are valuable aids to sort and retain all useful elements of best methods and to evolve from these a method worthy to be established as a standard and to be transferred and taught. Through this process is made possible the community conservation of measured details of experience which has revolutionized every industry that has availed itself of it.

Micro-motion study, presented for the first time at this meeting, is a new and accurate method of recording and transmitting skill. Based upon the principles of motion study and time study, it makes possible simultaneous measurement of both time and path of motions. It produces an entirely different result from any of the methods attempted by its predecessors, in that it shows a measured difference in the time of day on each and every cinematograph picture, even when the pictures are taken at a rate much faster than ever considered in work where positive films are printed and projected upon the screen.

The devices used in making micro-motion study are adaptable to the needs of the work. The kind of clock and camera used and the number of pictures taken per unit of time depend upon the nature of the work observed. For those interested primarily in the time study work of a machine shop, the clock that shows divisions of 1/200 of a minute is recommended; for ordinary problems of motion study, the clock showing divisions of 1/1000 of a minute is the best; and for those who desire to make the finest of motion and time studies for the purpose of obtaining ultimate methods of least waste, the clock showing divisions of $\frac{1}{1{,}000{,}000}$ of an hour or less is absolutely necessary. Such a clock is essential for discovering the method of least waste in cases such as handing instruments to a surgeon when operating. There is no case in the industries where the necessity for highest possible speed consistent with desired results is so great. For

example, when operating for mastoiditis, it is necessary that the probe to lift the scalp be used within the shortest possible time after the skin has been cut, before the blood has had time to run down into the cut. Micro-motion study and stereo-cycle graphs are the only methods that will measure the times and paths of different motion methods for doing this portion of the operation. Micro-motion study has already determined that the combining of two or more instruments or tools for such cases and reversing the ends in the hand is much quicker than dropping one tool and grasping another.

Not only is it possible with micro-motion study to make more accurate measurements of shorter times than one could with any other method of motion or time study, it is also less expensive, even for ordinary work, than the older stop-watch method. Much of the work can be done by a less skilful man than the old time-study man; moreover, provision is made for stopping all photographic expense during any time the worker is resting, or doing work where no record except elapsed time is wanted.

Recent improvements in the method of taking pictures and of using half-width films, with nearly twice as many pictures to the foot and about one-third as many to the second as is used in motion pictures of the standard "movies," have still further reduced the cost of taking micro-motion studies.

Because of the flexibility of the micro-motion study apparatus the possibilities of its use are much extended. It is possible to take pictures as slowly as is desired, for such observations for example, as are required on time study of the machine's time, such as one picture per minute, while when it is desired to study the minutiae of motions the pictures can be recorded at any desired speed, even at the rate of 1,000,000 per hour, for short periods. Recording as it does rest periods as well as work periods, micro-motion study presents complete as well as accurate records of the skill displayed.

These records are not only indispensable to those who are to teach or transfer the skill or experience, and are in themselves useful as object lessons, but more important than all else they are the devices that measure and record the skill which is to be transferred. They are used by the man who makes standards to determine the most efficient method of doing work. By them he is able to "take any motion apart" and to think in elementary motions. Being provided not only with a record of the best

method of the best man, but with records of the best methods of all most skilled workers, he can synthesize these into a standard method which will be better than any of the methods submitted, and is likely to be better than all combined.

To the worker this knowledge comes in various ways: He may be given photographic films depicting some method that he desires to acquire. These he can study at his leisure; making the demonstrator do the work as slowly as he pleases. The difficulty that most skilled workers find in making habitual motions slowly is a great hindrance to learning by observing them. The film records the swiftest motions, which can be taken apart and observed slowly. He may be given films on which are recorded methods of distant shops whose workers he would never otherwise observe. The worker may be taught not directly by the films, but by methods derived indirectly by them.

With records of skill that fulfil the three requirements of measurement, namely, (*a*) of proper units, (*b*) made by scientifically derived methods, (*c*) with devices that reduce expense, and with transference of skill that assures every worker an opportunity to acquire the best that has been thought and done in his line, scientific management can now look forward to fulfilling the ultimate demands, and justify itself from the economic viewpoint, and reduce the cost of the product to the consumer.

The Committee. The large amount of discussion offered on the report on The Present State of the Art of Industrial Management shows the interest in this subject on the part of members of the Society. This, and the manner in which the report was received, are sources of gratification to the committee. Although a few points were singled out for objection, the report as a whole seems to have been approved. In fact, these objections as they appear in the printed discussion, in most cases can be directly offset by quotations from the same source.

The first objection is that the report gives "no more than a fragmentary idea of the conditions under which this art is carried on in the United States today," first paragraph of Mr. Going's discussion. Offsetting this, we quote from the first paragraph of Mr. Gantt's discussion, "The committee have caught fully the present spirit of the movement now in progress, and Pars. 45 to 58 of their report seem to me to be an excellent resumé of the subject." It was obviously improper for the committee to consider the details of the systems of the art of man-

agement as practised, but it was essential to show the spirit of the movement and if possible the principles upon which it rests. This the report endeavored to do, and did do in the opinion of the member quoted above.

Mr. C. B. Thompson pointed out that there are several criticisms made by workmen against modern industrial management, and that labor unions are busily fighting its introduction. With further reference to these features is this sentence, "The report of such a committee as this should not have overlooked the opportunity to begin or extend the campaign of education in these particulars." And again, in regard to the dehumanizing effect on the workers of the methods of modern industrial management the same author says, "This committee must have had an opportunity to look into this side of the case; and it is to be regretted that they have not improved it more fully."

This author seems to have overlooked the necessity which compelled the committee to treat of their subject in a non-controversial manner. At the same time, the broad results from industrial management are clearly stated in Pars. 62 and 63. These seem sufficient to meet this criticism. In regard to the statement that the criticism of the dehumanizing effect should have been met, we need but quote from Mr. Coburn's discussion to show that the human side of the subject was found by one of the Society's members,—"the committee have expressed in their report the human interest side of scientific or labor-saving management, which some of its critics say it lacks." Further, the entire discussion of H. L. Gantt develops the point that the workers are benefited by the best of modern management.

The term, transference of skill, is used frequently in the report and conveys one of the important ideas in the definition of the new element in the art of management. This term seems to have been misinterpreted, for near the end of Mr. C. B. Thompson's discussion we read, "In this report the committee emphasize the 'transference of skill' as the basic feature of the new labor-saving management. Unfortunately, however, it appears that this term is used with two meanings. Throughout most of the report it seems to mean the accumulation of skill by the planning department and its transference from this department by actual instruction to the workmen just as machinery is said to be the transference of skill, according to the report, from the designer and draftsmen to the machine. The idea intended to

be conveyed is undoubtedly right, but the illustration chosen is unfortunate;" and in Mr. Vaughan's discussion, "the transference of skill referred to in the report means in one place, doing away with skill, in another the improvement of skill, and develops into the idea of telling men how to do everything."

The idea of the "transference of skill" is abstract, and these quotations indicate that their authors have failed to get the meaning of the term as used by the committee. "Transference of skill" is a process, and the expression might be expanded into "the process of transfering skill." It was the completion of this process that did away with hand looms and hand weaving in England. At the time of this completion, the skill in hand weaving was the personal possession of the last generation of hand weavers. Men of the succeeding generation did not acquire this skill for there was no economic advantage in so doing. Yet cloth was woven in greater quantities than before, leading at once to the question, where then was the skill? It is evident that the yarn was being manipulated by machines, thus, the former human skill was now in the metal fingers and arms and levers of the mechanism. But a new form of human skill was being developed as the process advanced; this is, the practical ability to tend the machine, keep it in order and producing to its maximum capacity.

The report pointed out that this process had advanced to a great length in the field of machine design, but had remained almost without application in the field of manufacturing. The best of modern industrial management applies this process to all of the activities of manufacturing, that is, a study is made of the steps of manufacturing in the same way that a study is made of the steps of designing. The results of handling and operation study are recorded for the instruction of the men engaged in manufacturing, in the same manner that the conclusions of the study of design are recorded on drawings as instructions for what is to be made. The parallel is exact.

It must be emphasized that this process applies to much more than the operation of machines, it applies to everything that is done in handling materials, machines, tools, and labor used in production. It is unnecessary to list these in detail, for everyone acquainted with manufacturing appreciates what is included. The training of the workmen is but one small, though important, part of the application of this process of transfering skill.

Regarding the discussion as a whole, there are two striking characteristics that attract and hold attention; the entire absence of exaggerated statement and the presence throughout of a humane spirit in keeping with the best trend of thought toward social justice. The first of these, the absence of exaggeration as to what industrial management has done or can do was to be expected in any discussion before a body of engineers. The second shows clearly the development that has taken place within the last few years leading to a new appreciation of the needs and rights of employees.

But this commendable attitude of justice does not seem to have had its full influence on the relations existing between management experts. Among some of these, there is an unfortunate spirit of intolerance. This is in marked contrast to the spirit prevailing in all the great divisions of engineering. In these there is room for the cadet engineer as well as the recognized expert and for many others between these two extremes possessing varied degrees of knowledge and experience, as witness the membership of our own Society. The same is true in the field of industrial management. There is room and need for everyone who understands the principles upon which it rests and who will conscientiously and intelligently apply them.

This situation and these facts should lead to a tolerant attitude between all who are honestly trying to further the art of industrial management, and there should exist the same spirit of mutual helpfulness and encouragement which actuates those in other lines of engineering specialization. On the other hand, all who are trying to exploit the present interest in this important subject for mere personal advantage must be unsparingly condemned.

No. 1379

THE PRINCIPLES OF VALUING PROPERTY

WITH SPECIAL REFERENCE TO INDUSTRIAL ENTERPRISES

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Member of the Society

In general every industry should be revalued at stated periods. The laws of some states require a revaluation every five years, or for purposes of assessment or insurance every seven years. In this paper the leading principles laid down for defining and ascertaining the value of property for various purposes are concisely expressed.

2 In discussing the several kinds of values, reference is made to a number of cases handed down from the Supreme Courts of several states and the United States, and some of the cases establishing the precedent upon which the rules of law are founded may be found in Judson's Taxation. In some cases the language used is that of the reported cases and in others of the textbook. The matter contained in the discussions of depreciation and capitalization is out of the author's own experience, with a brief reference to the report of the Tariff Board on the worsted and woolen industry.

DEFINITION OF TERMS

3 Following is a definition of the terms used, or referred to:

a *Cost of plant* includes the cost of the materials, labor, professional services required to produce the physical property, interest and insurance during construction, and the amount required to put the plant in operation.

b *Cost of operating* includes labor, supplies and incidentals necessary to manufacture the product, but does not include the stock in process, repairs and maintenance.

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c *Fixed charges* includes taxes, insurance, interest on current liabilities, and bonds.

d *Overhead charges* includes oversight, office and counting room expenses, advertising and selling.

e *Depreciation annuity*, the sum set aside as a sinking fund to replace the plant.

f *Annuity*, the sum set aside to discharge the funded debt at the end of a stated period.

g *Amortization*, the annuity set aside as a sinking fund to redeem the capital invested at the end of a stated period. The term refers particularly to the death of an industry from its exhaustion, as a mine or quarry. It is not depreciation, although analogous to it. Depreciation infers a continuance by renewals to the tangible property. The term amortization is not again referred to in the following discussions.

VALUE OF PROPERTY

4 In determining the value of property all the factors enumerated enter into the problem, and each must be given its proper amount of consideration and correct application to bear the test of logical reasoning. Every case of appraisal should be approached with the idea that the results may have to stand the test of the court.

5 Value does not mean cost, nor does it necessarily mean the purchase nor the selling price. It means the desirability or worth of one thing as compared with that of another, with reference to its usefulness. Value as defined by law is measured in terms of money.

6 A valuable consideration, however, within the meaning of the law may consist either in some right, interest, profit or benefit accruing to one party, or some forbearance, detriment, loss or responsibility given, suffered or undertaken by the other. It is under this definition that a court of equity would compel the specific performance of a contract where damages would not give adequate compensation as in a case where the thing contracted for is specific and its counterpart can not be purchased in the open market.

7 The rule of law would be to place a value upon the loss or non-performance of the promise and award the damages in a payment of money. Equity follows the law wherever possible,

and in a suit for damages where compensation can be given by the payment of a sum of money determined by the value of the property in question, equity would not grant specific performance, nor what is termed a payment in kind.

8 The theory of the law is that money as the universally recognized medium of exchange and measure of value, is always acceptable and leaves the one who rightfully possesses it free to purchase what he wishes to satisfy his requirements or pleasure. It is very rare, indeed, that a demand or pleasure can not be satisfied by the purchase of something else.

9 Another reason for preferring a payment of damages in money is that upon the payment of money litigation ceases; whereas, if specific performance, or payment in kind were granted, the courts would have the difficulty of enforcing the specific performance, or payment in kind.

10 Payment in kind, in the case of water rights, has been allowed in England, by act of parliament.

TAX VALUE

11 Tax value is defined by the courts and stated by the legislators of the several states to be the true or fair cash value of the property at the time of assessment. The fair cash value is the price which a willing and voluntary seller will accept from a willing purchaser who has the money to pay at the time of sale. In the assessments of taxes the rates will vary from nothing in the case of exemption to over $20 per thousand. Some localities may have a low rate of valuation with a low rate of tax, others may have both a high rate of valuation with a high rate of tax, while others may have a moderate rate for both valuation and tax. There is no violation of the law as long as the rates are uniform and equal. The United States Supreme Court laid down the rule that uniform and equal refers to geographical uniformity where the rates operate generally throughout the United States, and uniformly wherever the subject of the tax is found. The court has declined to define the word equal as no rule can be formulated to cover every case, but upon the general question it is said that equal protection of the law means that no person or class of persons shall be denied the same protection of the laws enjoyed by other persons or other classes of persons in the same place and in like circumstances.

12 Some of the states hold that equal and uniform mean in-

trinsic equality and uniformity, which is not apparent equality or uniformity. The Supreme Court of Massachusetts said that a fair cash value is irrespective of value placed by the assessors upon similar property in the same city, the question being whether it had been valued at more or less than its fair cash value, and not relatively more or less than similar property of other persons.

13 The amount of taxes assessed is usually taken from about two-thirds to three-quarters of the full value of the property, but this is not always the case. While making an appraisal recently the author found that the rule of the locality was to collect the amount of tax upon 80 per cent of the full value, while the tax rate was $18 per thousand upon a fair valuation.

14 It is obviously immaterial what the basis of valuation is if it is uniform as to all property within the territory, or as to the class of subjects upon which the tax is laid. This is recognized in the requirements of state constitutions that taxation shall be uniform upon the same class of subjects within the territorial limits of the authority imposing it. Thus, if all the property in the state were valued on the same basis it would be immaterial to the individual tax payer whether he paid 1 per cent on a valuation of $1, or 2 per cent on a valuation of 50 cents. If there was no general property tax levied by the state, based upon valuation, it would make no difference whether property in one town or county was valued on a higher basis than property in another. But within the territory wherein the tax is levied, inequality of taxation results as certainly from inequality of valuation as from inequality in tax rate. The failure to recognize this fundamental principle in taxation can result only in misleading comparisons.

15 It has been held by the courts that the evidence of the valuation of other properties is admissible as a possible assistance in determining the cash value of the property in question, when the cash value and not the proportionate value is the point in issue.

16 Stock market quotations of the company's securities may be considered because the stock and indebtedness represent the property. But they are not to be regarded as conclusive standards or tests of value. In the language of the United States Supreme Court such quotations represent the faith which a purchaser of stock in such a company has in the ability with which

the company will be managed, and in the capacity to make future earnings. Whether it is well or ill founded, is but a matter of opinion. While the value of the property is one of the material facts going to make up the market value of the stock, it is not the sole one. Speculation has been known to exercise a potent influence on the market price of stock. But assessors can use this as evidence of value.

17 Capitalized net earnings as a means to determine the fair cash value of public utility, or quasi-public corporations, including the stock, franchise and tangible property, have been allowed by the court. The basis of the valuation was taken as the net earnings for the year, which did not appear to be an exceptional one, and capitalized at 6 per cent; this ratio being adopted in a special case, is the rate adopted in states where assessments were made upon the basis of net earnings.

18 The court also said that this rate was less than some advanced advocates of municipal ownership were willing to guarantee to investors in securities of this character, but it was deemed justifiable by the considerations that attended the real investor's purchase of stock. To the value of the stock and franchises determined by capitalization of net earnings was added the value of the tangible property. In a case where there were no net earnings, but an apparent loss, the court said there was no basis upon which to assess the stock above the value of the tangible property.

INSURANCE VALUE

19 In cases of personal property the measure of damages for loss of the insured property is its market value at the time and place where the loss occurred, within the amount named in the policy. In the case of damage to an insured building, the rule is indemnification to the owner, not exceeding the sum insured. The question is not what someone would have paid for the building, but what amount will indemnify the insured for the loss sustained.

20 Where goods are insured the insurer is bound to pay their value at the time of loss; where damaged only he is bound to pay the difference between the value in the sound and damaged condition. Market value, and not local or peculiar value must control.

21 In the case of the destruction of a building, where the policy

is not valued, the damage would be the value of the property as it stood upon the ground the day it was destroyed. If the building was old and dilapidated by use and decay, its value in that condition should be the ground of recovery. If the policy stipulates that the cash value of the property destroyed or damaged shall not exceed the cost of rebuilding it, and in case of damages from use or otherwise that a suitable deduction shall be made from the cost of repair, the measure of damages is the cost of repairs, if thereby the property is rendered as valuable as before. If less valuable than before, then the difference must be added to the cost; if more valuable, it must be deducted.

22 Under the laws of Massachusetts, no insurance company shall knowingly issue any fire insurance policy upon property within the Commonwealth for an amount which, with any existing insurance thereon, exceeds the fair cash value of the property, nor for a longer term than seven years. If the buildings insured against loss by fire are totally destroyed by fire, the company shall not be liable beyond the actual value of the insured property at the time of the loss or damage.

FAIR CASH VALUE

23 Fair cash value is defined as being the price a willing seller would accept from a willing purchaser, without any undue advantage to either party.

24 The influences which affect the salable values of property are variable and often complicated. It has been said that differences between assessors on questions of valuation of the same class of property are no greater than frequently arise between witnesses in a trial on questions of value. There is no certain, definite standard of values excepting that of money and standard marketable articles. Many influences, tangible and intangible, affect the salable value of property, real and personal, both in city and country, so as to make its real valuation a work of great difficulty, resulting in inevitable inequalities. Mere mistake in judgment is no discrimination.

25 The Massachusetts rule is that the fair cash value is determined upon the basis of a willing seller and a willing purchaser.

26 In New Jersey the rule has been stated as being the ordinary selling and buying price for cash at the time, irrespective of the actual cost of the property, the amount it would realize at a public sale being a good criterion. This rule would seem to

admit of the possibility of some undue advantage to the purchaser, as a public sale might not always be a willing sale on the part of the vendor. On the other hand, the reason for the rule suggesting public sale as a criterion for establishing values is undoubtedly based upon the possibility of inflating values for some reasons, or discounting them for other reasons; also for the purpose of lending aid to a uniform and equal basis of valuation on the ground that the records of such sales of property of the same class, within the territorial limits, will be available to form a standard of comparison.

27 The Massachusetts rule employing the words willing seller and willing purchaser implies the status of equality between the parties which is not in the nature of a forced sale. In a case where a manufacturing property was sold at public auction, on account of the owner's personal wish to sell the property, there is the fact of a willing sale to the purchasers, who paid about one-quarter of the appraised value of the property, as tangible property. If the sale had been made under legal process, by order of the court, it would have been a forced sale, even though the purchase price paid was the same. The price paid under the circumstances may be held as not representing the fair cash value. The sale can be said to resemble more closely a sacrifice by the owner.

COMMERCIAL VALUE

28 The commercial value is the estimated worth of property placed upon it as a business proposition, the chief considerations being the earning capacity of the plant and the amount of profit to be derived from it, both present and prospective. Involved in these considerations are location, facilities for receiving and shipping goods, available supply of help, management and reputation.

29 This value is to be determined by a proper capitalization of the earnings, which for manufacturing properties will fairly represent the market value of the property. It is evident that, as to the owners of a prosperous industry, the commercial value may be above the market value of property of the same class, also that some industry may be below the market value of property of the same class. The method of management may have the weight of influence to determine the commercial value in some cases, or a peculiarly favorable market for the product. These reasons are largely of local value rather than of general value. In the case

of poor management there is a factor that can usually be remedied. In the case of a peculiarly favorable market, the influence of competition, or the changes in fashion, will in time level the particular advantages accruing to the individual plant.

30 Industries may be so located as to involve the expenditure of a larger amount of money in the establishment of their tangible property than other plants more favorably located, owing to some particular advantage gained at the time of their inception. With the advent of new inventions and improved methods, the reason for having the plant in such a locality has passed away. Conditions and environment may result in larger operating expenses reducing the profit. As a business proposition the value of the tangible property can be no greater than that of other plants of the same class more favorably located and costing less; and their commercial value should be determined by an amount equivalent to the charge upon the profits owing to the unfavorable conditions. This is illustrated by many cases where plants have been located in out-of-the-way places to take advantage of some water power. With the advent and increased use of electricity, with the growth of the plant beyond the capacity of the water power, with a growing scarcity of help, and with a more favorable locus for producing the same kind of goods in other places, these plants have been forced to suffer loss of money and prestige.

31 In valuing industrial property for tax assessment, insurance indemnity or for sale, the commercial value is the leading consideration, as upon it would depend the attitude of the purchaser towards the seller. In determining the value of property upon the basis of net earnings or profits the amount to charge off against the tangible property would be the extra expenses involved in manufacturing the product that could be saved if the plant was working under favorable circumstances. This would not necessarily affect the value of the collateral intangible or incorporeal property, except in so far as the loss of profit might affect the sale of the stock, which might also be affected by loss of profit due to poor management. While poor management may have a very material affect upon the commercial value of an industry, it is subject to correction, and the result of poor management may be felt for some time after the cause has been removed. It is not inherent in the property the same as unfavorable location, the betterment of which would mean the removal of the plant.

32 In valuing the tangible property, its adaptability for other uses is to be considered. This involves the amount of capital which must be expended to put the property into a proper condition to carry on the business for which it may be purchased. In the event of carrying on the same class of business the question of the economical organization of the plant, as compared with other well arranged plants entering into competition would be involved.

33 The uniform and equal valuation of textile, and other industrial property and power plants should by this time be well within the range of possibility, and no one should be any better able to make fair and unprejudiced appraisals than the mechanical or industrial engineer. There is a sufficient amount of information at hand true to the facts of cost, both for installing plants and for operating them, covering all classes, especially of the textile industries. The engineer, who has made certain industries a specialty in all of the processes of mechanical organization, cost of establishing, and cost of operation, should be better qualified than any other man to act as an expert in such valuations. He has a broad field from which to gather his information and a larger opportunity to view the various grades of the same classes of industry. The ethics of his profession should lead him to be an unprejudiced witness before the court. The theory upon which the expert witness is allowed to testify is that he can assist the court to arrive at a just decision between the parties in a controversy, owing to his superior knowledge and experiences. His evidence should be free from the taint of partisanship, and he should stand before the court in the same attitude as an attorney, representing his client, but serving the court. The expert witness is the only one allowed to give opinion evidence, on the theory of his special qualification.

34 No valuation can be made without taking into account the depreciation.

DEPRECIATION

35 Depreciation is the falling off in value. Physical depreciation refers particularly to the lessening value of material things, due to age and wear, and improvements in the mechanical arts. Everything of substance has some intrinsic value, however small.

36 There are several methods of computing depreciation;

some are represented diagrammatically by curved lines, others by straight lines, all of which methods show some variations, but any method adopted may be only approximately correct. Depreciation begins at once, however small, and the actual process will not proceed very far before the work of maintenance begins. Whatever the method or rate of computing the depreciation, if it is sufficient to meet the conditions its relative accuracy is near enough to the truth for all practical purposes. Then it is logical to utilize the simplest form of diagram to represent the loss, which is the straight line computed on the original cost, or the replacement cost if new at the time of making a valuation of property.

37 While it is not the purpose to discuss the several methods of computing the depreciation here, as these discussions may be read elsewhere, the rate of depreciation may be briefly considered.

38 While the rate of depreciation will vary with materials and circumstances, it may be possible to determine a fairly constant rate applicable to property of the same class, used under similar conditions and subject to the same general care and maintenance. In the case of the Chicago Union Traction Company vs. State Board of Equalization, 114 Fed. Rep. 557, it was held that an annual reduction equal to 6 per cent of the current value of cars, tracks and machinery was not excessive as the allowance for current depreciation.

39 In the textile industries, plants may be grouped together in classes, each class being subject to the same economic conditions. In the cotton industry, plants may be classified as duck mills, course sheeting mills based on a standard weight of cloth containing three yards to the pound, medium weight goods mills based on a standard of four yards of cloth to the pound, print goods mills, making goods for the general household trade, and fine goods mills. The mills of each class enter the market to sell their product in competition with the other mills of the same class. Each mill of the same class must strive for the highest efficiency in order to compete successfully with the others, and these conditions have led to a highly scientific development of the industry. In the woolen and worsted trade, the same highly efficient physical organization is being wrought out, until it seems as if there can not be any very radical changes in the methods or general types of machines used in the process of manufacture.

40 In all mechanical and industrial processes the same laws of

growth and development are as potent as are the laws of evolution in the natural growth and development of the physical world. The very nature of the materials we have had to convert into a marketable product for our use has indicated the lines of development for the mechanisms of our industrial processes. Many improvements are yet to be made and perfected.

41 One of the most notable developments and advancements in the art in recent years is the advent and successful operation of the steam turbine, and prior to it the introduction and development of electricity for all purposes. One of the greatest developments yet to come is the method of generating heat without coal.

42 A statement was recently made that no great advancement has been made in the cotton industry for about forty years. This is a cause for surprise and some inquiry. The period from 1870 to 1880 appears to mark the beginning in improved modern buildings, better lighted rooms, better fire resisting construction, more economical arrangements of plants generally. The decade from 1880 to 1890 appears to have been the period of increase of the product of machinery and of a higher efficiency of the plant generally. The decade from 1890 to 1900 appears to have been the time for introducing better organized plants, of giving more attention to the welfare of the employees and tending to raise the standard of quality generally. The last decade has witnessed the introduction and successful operation of electricity generally, better appointed power plants, more efficient machinery and buildings, and the study into the question of more efficient work people.

43 In the decade from 1870 to 1880 the average speed of the front roll of spinning frames on print goods numbers was 64 r.p.m., the average product being about 0.80 lb. in 60 hours. Today the average speed of the front roll for the same numbers of yarn is 122 r.p.m., with the average product of 1.44 lb. per spindle in 60 hours, an average increase of 80 per cent per spindle, with an infinitely better quality of yarn. The product of looms has increased from an average of about 27 lb. per week to an average of 44 lb. per week on print goods. Forty years ago the average product per week gross floor area was about 1 lb. of cloth to 7.25 sq. ft., while today in modern mills the average product is about 1 lb. of cloth per week to 3.5 sq. ft. for a week of 60 hours. These results have been calculated from the records of a number of mills of the earlier period and compared with what is seen in practice today. The increased product of machinery may show

a greater ratio than the increase per square foot of floor area for the reason that in our modern mills more alley room is allowed around the machinery, spinning frames are wider and built to a larger gage than in the earlier days, looms are larger and heavier for the same width of cloth. There is a saving in the card room, as the revolving flat cards used today occupy less room than the double system of flat top cards used years ago, and a large increase in weaving, due to the use of automatic looms.

44 Within the last ten years there have been some marked improvements in the building of machinery. It is built more strongly, on better lines, and is more rugged. This is quite apparent in the case of some steam engines, as compared with earlier types.

45 As to the depreciation rate to charge off against machinery in the case of textile machinery, it is a common thing to find groups of the same kind of machines in a mill varying in age from fifteen to eighteen years; sometimes one sees a group of machines over twenty years old, and it is not uncommon to see single machines from thirty-six to forty years old. As a rule it would be beneficial to the organization to dispense with these old machines. The writer visited a mill sometime ago where there were some old roving frames and was told by the agent that it was more economical to let the old frames remain idle than to try to run them. It is short-sighted policy that keeps such old machines out of the junk heap.

46 By the time spinning frames reach the age of fifteen years they usually begin to require more extensive overhauling than ordinary repairs. The writer has known frames of about this age to require such thorough refitting that the standards were about all that was left of the original machine.

47 During the recent investigation by the tariff board appointed by the President of the United States, some rather interesting data were obtained regarding length of time machinery has been used in worsted and woolen mills, not necessarily indicative of the rate of depreciation of such machinery, although certain inferences may be drawn. Some of these data are given in Tables 1-5.

48 There is nothing in any of the tables given to show the age of any of the individual plants from which these lists were made up, so that no allowance can be made for new industries starting up within the last ten years, but between the ten and twenty year period there is about the same ratio for each period of five years.

and after this period the amount of machinery that has been in operation is relatively small.

49 From personal observation among a large number of textile mills the average rate of depreciation appears to follow along

TABLE 1 YEARS IN OPERATION OF WORSTED MACHINERY, BRADFORD SYSTEM

	Scouring		Carding		Combing		Gilling		Drawing	
	No.	Per Cent	No.	Per Cent	No.	Per Cent	No.	Per Cent	No.	Per Cent
Less than 5 years	11	16.4	173	26.3	53	14.0	137	18.5	797	32.9
5 years but less than 10 years	26	38.8	143	21.7	136	35.8	214	29.0	628	26.0
10 years but less than 15 years	17	25.3	76	11.5	18	4.7	68	9.2	126	5.2
15 years but less than 20 years	5	7.5	61	9.3	46	12.1	66	8.9	434	17.9
20 years but less than 25 years	5	7.5	155	23.6	43	11.3	116	15.7	92	3.8
25 years but less than 30 years	2	3.0	31	4.7	47	12.4	71	9.6	214	8.9
30 years and over	1	1.5	19	2.9	37	9.7	67	9.1	129	5.3
Total machines	67	100	658	100	380	100	739	100	2420	100

TABLE 2 YEARS IN OPERATION OF WORSTED MACHINERY, SPINNING AND WEAVING

	Mules		Frames		Looms*	
	No.	Per Cent	No.	Per Cent	No.	Per Cent
Less than 5 years	92	24.9	273	20.3	4125	33.4
5 years but less than 10 years	121	32.7	559	41.5	1244	10.0
10 years but less than 15 years	12	3.2	109	8.1	2009	16.3
15 years but less than 20 years	34	9.2	76	5.6	2072	16.8
20 years but less than 25 years	100	27.0	98	7.3	2053	16.7
25 years but less than 30 years	11	3.0	116	8.6	396	3.2
30 years and over	0	0	115	8.6	438	3.6
Total machines	370	100	1346	100	12,337	100

* Includes all looms for both worsted and woolen mills.

these general lines. Up to about fourteen years of use the machinery contains practically the original parts with which it was furnished, subject to the ordinary repairs that an economical mill will keep up. Of course accidents or some unlooked-for event

TABLE 3 YEARS IN OPERATION OF WOOLEN MACHINERY

	Cards		Mules	
	No.	Per Cent	No.	Per Cent
Less than 5 years	40	9.6	47	8.0
5 years but less than 10 years	62	14.8	204	34.7
10 years but less than 15 years	68	16.2	143	24.3
15 years but less than 20 years	16	3.8	72	12.3
20 years but less than 25 years	36	8.6	47	8.0
25 years but less than 30 years	32	7.6	53	9.0
30 years and over	165	39.4	22	3.7
Total machines	419	100	588	100

TABLE 4 RELATION BETWEEN DOMESTIC AND FOREIGN MADE MACHINERY, GIVING THE NUMBER OF MACHINES IN PLANTS LISTED

Country	Scouring	Cards	Combs	Gills	Drawing	Spinning	Looms
United States	37	730	57	37	273	617	9517
England	18	271	323	210	2488	1486	2713
Germany	7	90	225	90	192	195	107
France	5	0	22	32	36	6	....
Total	67	1091	627	369	2989	2304	12337

TABLE 5 AVERAGE YEARS IN OPERATION OF WORSTED MACHINERY LISTED IN TABLES 1 AND 2

Years in Operation	Average Per Cent of Total Machinery
Less than 5 years	26
5 years but less than 10 years	29
10 years but less than 15 years	10
15 years but less than 20 years	11
20 years but less than 25 years	13
25 years but less than 30 years	6
30 years and over	5

may happen to a machine to cause some extraordinary expense. From the tenth to the fifteenth year there is likely to be some very general overhauling of the machinery, renewal of vital parts, and perhaps the substitution of one or more machines in a group of similar machines. The whole organization is toned up generally. Conditions will run along for the next ten years or so under the ordinary mill care when there begins to be a very general reorganization of the plant.

50 The inference might be drawn from Tables 1-5 that the general, or average, depreciation may follow the results shown, and this does not seem an unreasonable, although it may not be the true conclusion. Assuming that the inference is approximately true, it seems that a depreciation annuity of 6 per cent on the cost of the machinery is not too much to set aside in order to provide a proper and conservative amount with which to replace textile machinery. This would allow for a partial replacement at the end of the 15 year period, and a general renewal of the plant at the end of 25 years. If the plant is in a good condition at the end of 25 years and can continue to operate for a few years longer, the gain is so much greater.

51 Industrial buildings are planned for certain specific purposes. The organization of the plant is first worked out, the departments arranged most economically, then the walls and roof put over the whole.

52 Under proper conditions, buildings will last for hundreds of years, but with factory buildings, the conditions are such that it is not expected to have a building last much beyond a certain limited time, owing to the change in methods. It is noticeable, however, that the buildings for manufacturing purposes of recent years are more substantially built than they were 50 years ago.

53 In looking over the records of existing textile industries, buildings in use which are over 50 years old are not very numerous; there is a larger number of those 40 years old, and a rapidly increasing number in comparison as the age diminishes. The old buildings are wholly unsuited for modern practice. In fact, many buildings built 25 years ago are not suitable for a proper arrangement of modern machinery.

54 With more rugged and better built machinery and stronger and better made buildings it is possible that the depreciation will not be so rapid as it has been in the past. To a very large extent

depreciation is due more to rapid advance in the mechanical arts than to age and wear. The proper maintenance of a plant can retard the depreciation due to age and wear very materially. Depreciation due to advance in the arts is, in the author's opinion, one of the essential factors to be considered in the use of reinforced concrete buildings, as this type of building is not so readily adapted to be modified or changed over to meet new conditions as is the regular mill construction type. It may be, however, that radical changes are not so likely to come in the future as they have been in the past, but this seems a short-sighted view.

55 In depreciating factory buildings, a rate of about 2 per cent a year does not seem excessive, compared with the useful life of the older type of building, while about 1 per cent a year may be large enough to allow for the depreciation of the modern buildings.

56 In making up valuations of textile plants the physical depreciation for the preparatory, spinning and weaving machinery should be taken at about 4 per cent a year, while for dye-house and wet-finishing machinery the wooden parts of machinery have to be entirely renewed on an average of every five years, so that the depreciation on these machines after three years is very nearly a constant rate for a going concern, and can be taken at about 25 per cent of the replacement cost.

CAPITALIZATION

57 To capitalize is to convert an income, annuity, etc., into a single payment or an equivalent capital sum.

58 Under the subject of capitalization we must take into account what is included in the fixed charges on the plant. It is a very easy thing in handling capitalized values not to include some element in the calculations that belongs there and so raise false values, or vice versa.

59 The fixed charges usually include taxes, insurance and interest on current liabilities. The factors subject to capitalization are the annuities for depreciation, the annuities to discharge the funded debt and a fair profit to be derived from the enterprise.

60 The rate of capitalization is not always easily determined and may vary under varying circumstances for the same class of property.

61 In making up the commercial valuation of the railway

operating property of the United States in 1904 the rates of capitalization vary from about 3½ per cent to about 10 per cent based on the net earnings of the railroads, and the total of these capital amounts represented the wealth of the railroad property in this country at that time. The whole subject is discussed in Bulletin 21, Bureau of the Census.

62 In determining the rate of capitalization of values for industrial properties, Table 6 shows the factors which may be taken. The table is based on the assumption that the economic life of buildings is 50 years, and that the annuities are invested to earn a return of 4 per cent compound interest, the buildings

TABLE 6 FACTORS IN DETERMINING RATE OF CAPITALIZATION

	Annuities	Percentage of Contributions		
1	Buildings depreciated at 2% a year	0.25	0.25	0.25
2	Machinery depreciated at 4% a year	1.50	....	....
3	Machinery depreciated at 5% a year	....	2.10	....
4	Machinery depreciated at 6% a year	....	....	2.65
5	Annuity to discharge funded debt, say	1.12	1.12	1.12
6	Fair net profit	10.00	10.00	10.00
	Total	12.87	13.47	14.02

being about three-eighths and the machinery about five-eighths of the cost of the tangible property.

63 As the plant grows old and improvements are made in methods and efficiency the net profit will decrease. Assuming this decrease to follow at about the same rate as the average physical depreciation of the property, there will come a time when it will cease, unless proper precaution is taken to provide a fund large enough to maintain the property by renewals from time to time; and the time will always come when some general reorganizing and renewals are necessary. It seems justifiable to take the depreciation fund at 6 per cent per year of the cost for textile machinery, and following similar periods of usefulness to allow for not only the renewals which must be provided for in due course, but for renewals due to improved and more economical machinery.

64 A gross income of 20 per cent should be conservatively assured from a new industry not only to cover the annuity

charges and the profit expected to be realized from year to year, but also the extra expenses of establishing the plant and the risk involved in floating the enterprise.

65 In capitalizing values the rate of 10 per cent is often used. If this includes the annuity charges for depreciation, then the allowance for the net profit is only about 5 per cent. If it is assumed to include the net profit only, it is a more conservative assumption.

66 The method of capitalizing a possible saving to an industry to determine comparative values is often not correct, as the capitalized values lead to exaggerated conclusions which are wrong. What percentage of saving could be effected by comparing the costs of installation and operation between two processes should be ascertained, and the values then determined from the cost of installation. If the saving amounted to 15 per cent, the cost of making a change would be considered justifiable; if to 10 per cent, the cost of making the change would not be considered justifiable, except as modified by the question of necessity, as by the renewal of an old plant under similar conditions. A saving of 10 per cent might warrant a change in methods. Thus in ascertaining comparative values by the method of capitalizing possible savings, it would seem as logical to use a rate of 15 per cent on the basis of the above reasoning.

67 A low rate of capitalization may lead to exaggerated and wrong conclusions. It is easy to mistake the principle involved, and rate the value of tangible property on the negative side of the equation. A small variation in the rate results in large variations in the values. The values for tangible property should always appear on the positive side, as all material property has some intrinsic value, however small.

68 As an example of this principle we will assume a cotton mill equipped with 500 automatic looms and 500 plain looms. The plant is making a class of goods where automatic looms can be used to advantage. The product of the automatic looms is about 45 lb. of cloth per week per loom, that of the plain looms about 40 lb. of cloth per week per loom, or say about 1,000,000 lb. per year for the 500 plain looms.

69 By using 444 automatic looms in place of the 500 plain looms a net saving of about 1¼ cents per lb. of product can be made, or a total yearly saving of $1{,}000{,}000 \times 0.0125 = \$12{,}500$. The cost of 444 automatic looms at $168 each, including installa-

tion, would be $74,592, and if we allow a salvage value of $10 per loom for the plain looms the net cost of installing the automatic looms would be $69,592. A yearly saving of $12,500 would be about 18 per cent of the cost of installing automatic looms, so that the mill would be warranted in making the change.

70 The replacement cost of 500 plain looms, if new, would be about $30,000. If we capitalize the saving of $12,500 in favor of automatic looms at the rate of 15 per cent, the capital value equals $83,333. We have already allowed a salvage value of $5000 for the plain looms so it is evident that the only value of the capitalized saving in favor of automatic looms to be applied to the plain looms is an amount having the positive value of $5000.

71 Again, assume that the cost of installing the automatic looms is $74,592. The replacement cost of the plain looms if new is about $30,000, which depreciated for age and wear is at present $5000, with no rebate by the loom company as an inducement for taking these looms out and replacing them with automatic looms. The gross cost chargeable to installing automatic looms would be about $79,592, including the value of the plain loom, and a yearly saving of $12,500 would be about 16 per cent of this cost in favor of automatic looms, which indicates that the investment would be warranted.

72 With reference to the commercial value of the plain looms, the capital value of the saving that could be made by using automatic looms can not be applied to reduce the value of the plain looms by an amount equal to more than their depreciated replacement cost, as a positive value, or $5000.

73 One of the hardest experiences is to recuperate an old plant after it has been allowed to reach such a state of physical depreciation as to become profitless, and there has been no depreciation fund provided to take care of future contingencies. Such conditions are probably more likely to occur in privately owned industries than in corporations. There are numbers of cases where plants in such a condition have been taken over and operated, but this means the sinking of new capital.

74 A case coming under the author's observation is that of an old mill which had reached the limit of its usefulness as a producer. Under a proper management, the old buildings were replaced by a substantial modern mill. During a period of depression the conclusions as to the wisdom of building a new mill

were that with the old buildings the plant would have been forced to close, owing to the stress of circumstances. The light was so much better in the new mill that the goods formerly passing as first quality in the old mill were not equal to the second quality in the new mill. Owing to trade conditions, it had been necessary to cut wages of the operatives twice, yet with the reduced wages the best operatives were able to earn more money in the new mill than they had been able to earn in the old mill before any reduction in wages had been made.

METHOD OF VALUING A PLANT

75 In the event of making a valuation of property, the first essential thing is for the appraiser to go over the property as a whole, to see how the plant is arranged, and how the arrangement compares with other plants making the same general kind of manufactured product. This survey will usually disclose any special features that may be unfavorable to a more economical arrangement, or where certain alterations might be made with benefit to the plant, as the organization and arrangement should be considered one of the vital features in ascertaining values.

76 It is not enough to say that the physical property of an industry can be replaced for a certain sum of money, or that it costs a certain amount and is to be valued at a certain other sum of money after allowing for a certain depreciation due to age, wear, obsolescence, etc. An investigation must be made into the working of the plant, comparing it with other plants in competition with it. Its location must also be considered.

77 Any extra cost of operation that can be eliminated by a proper reorganization must be accounted for and set off against the value. Sometimes the cost of reorganizing a portion of the plant is greater than the expense of making the change would warrant, but the poor feature will remain nevertheless a detriment to the plant and affect the value.

78 Inventories of property can be given by the owners of the plant, but these inventories should always be checked up by a person familiar with the business, who knows the process of manufacturing the goods produced, the machine used in each process, and how they should be arranged in each department, and who is capable of taking a birdseye view of the whole property and seeing it as a working whole, as well as segregating the several parts.

79 For this reason the engineer who is familiar with certain lines of industry is better able in general to take an unprejudiced as well as a more comprehensive view of the subject, owing to his wider experience and more special knowledge of engineering than a person engaged in commercial lines.

DISCUSSION

Charles H. Fish. The question of the valuation of property, speaking particularly of manufacturing property, is one which, unfortunately, can be looked at from a number of points of view, all of which are more or less reasonable. The answer to an important question always asked as to the age of a machine, a group of machines, or a building, is apt to be misleading. There are a number of kinds of machines which have been but little improved upon during the past period of years, and on account of this slow advance in the state of the art, the age alone is of comparatively small importance. Again, the question as to how carefully or intelligently the repairs have been kept up is one that must be considered in determining the present value. To quote from Mr. Rowell, Par. 46, "The writer has known frames of about this age (15 years) to require such thorough fitting that the standards were about all that there were left to the original machine." In other words, when the repairs were finished, these frames had new roll stands, new rolls, new spindles and rings, or in fact all of the important operative parts of the machine had been renewed. Admitting that the design of these machines was good, as no doubt it was, did they not, by these repairs, practically wipe out 15 years of age with which these frames were charged, bringing them down somewhere near a new frame age?

At the same time, in a recent experience with an association of insurance companies, the writer found it very difficult to get them to depart from the old theory that all machines have a certain, more or less definite life, at the end of which they were practically dead and worth only their value as scrap, and following this theory, they claimed that a machine, the life of which was called 20 years, was half worn out at the end of ten years. There are many machines in textile mills that are of such design or their operation of such character, that their active life can, by intelligent repairs, be continued almost indefinitely. These machines may die at the end of 20 years if allowed to by neglect-

ing repairs, but as a rule they are well, or at least fairly well, kept up. On the other hand, it is a fact that some machines are bound to wear out in spite of attempted repairs, or many become worthless owing to new improvements and inventions, but to state that the life of a machine or a building is a stated period of time, or that its value depreciates at a definite percentage per year, is very often wrong.

One of the principles of valuing property should be a careful detail investigation of each part or portion of the property in order to determine its actual physical condition and not to be guided too much by the one factor "age."

In reply Mr. Rowell said it was his intention to bring out here what might be called secondary depreciation. At the end of a period a machine, or group of machines, might be revived so as to be comparatively new, yet they were not as good as wholly new modern machines would be on account of improvements made in patterns, design, or what has been called advancement in the art.

C. J. H. Woodbury said that the distribution of power by various types of porches following the open belt porch, was an important feature in depreciation of mill property. Porches enclosed by wood sheathing, followed by the type at the side of the building and others, in all of which power was distributed along the lines of belting, and finally the belt porch projecting into the mill by which the power was distributed along the lines of shafting, were an indirect source of depreciation. In new mills these forms of construction had been displaced by the electrical transmission of power, which did not require more than a provision for the conductors. The old mill buildings were depreciated by this subsequent electrical invention because these belt porches formed a portion of the structure and for reasons of strength could not be removed.

Thus the advent of the electrical transmission of power with its freedom from requiring structural provision in a mill hitherto necessary, had within a few years enabled engineers to design a mill solely as a manufacturing structure without making arrangements for belt porches, without their attendant expensive impediments to floor space and light, and this improvement has in like manner depreciated mills of earlier types of construction, independent of wear and tear.

There was a depreciation from the contingency of an abandonment of the site of the mill, especially as the result of a fire. In such a case the owners of a mill at a water power distant from a railroad might find it more advantageous to connect the power house to an electrical generating plant and carry the wires to another site along the line of a railroad, than to rebuild on the old site.

A saving in gas of $3000, or 6 per cent, on the cost of a mill, was revealed in the case of removal of the machinery to a new building having larger windows. In both instances the buildings were lighted by the town gas at the same rate and the identical meter was used. It would not generally be assumed that modern windows would depreciate a mill having old types of windows 6 per cent.

He said another form of the distribution of depreciation was that of insurance. He knew of one cotton mill which had been insured for a number of years, under the factory mutual insurance plan on the basis of paying gross premiums subject to returned dividends, which instead of being credited to the insurance account, were invested in the stock of the mill purchased on the open market and in a few years a permanent insurance fund thereby secured, the income of which in the shape of dividends was sufficient to pay for the insurance without annual expenditures from the merchandizing income of the mill.

Charles T. Plunkett asked if the author's statements "depreciation is the falling off in value," and "depreciation begins at once, however small," must be taken in with a series of years in order to make up the depreciation, since much machinery improves during the first year. He was answered in the affirmative.

From the author's statement regarding the making up of the valuations of a plant, "an investigation must be made into the working of the plant, comparing it with other plants in competition with it," Mr. Plunkett wanted to know if the valuations were comparative. The author said he regarded commercial valuations as comparative.

The Author. The discussion of the subject has turned upon the matter of depreciation which is quite a difficult problem. It seems that instead of closing the subject with this paper and its discussion an effort should be made on the part of the Society to take up the subject of depreciation and maintenance to de-

termine if possible what may be the best methods to pursue in various industries and so leave the question open for further enlightenment.

No. 1380

TRAIN LIGHTING

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The first railway trains, usually made up of one car, made short trips in the day time only and artificial light was not required; as the trips were extended into the evening the cars were still unlighted. The lighting of stage coaches before the day of railway coaches was obtained by candles brought in by the passengers themselves and placed in sockets along the sides of the coach. It is recorded that the first railway car illumination can be traced to one Thomas Dixon, a driver on a horse train railway in England in 1825, who out of his generosity placed a penny candle on a table in the center of the car for the convenience of his passengers. The early railroad patrons who contemplated traveling by night, and were foresighted, brought their own candles; and later the candles were supplied by the railroad companies in crude sockets, but the passenger had to light them and nurse the flame.

2 A review of the development of railway car lighting shows that it is divided into periods of approximately twenty-five years for each of the illuminants used, beginning with candles in 1825. In 1850 oil lamps were introduced, which gave way to gas in 1875, and this in turn was superseded by electricity in 1900. The development of each of these systems extended over a number of years and the new illuminant did not entirely displace the previous method of lighting. In fact candles are

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sometimes used today for emergency lighting, while oil lamps may be found in many coaches. With the early applications of electric lighting to cars it was usual to retain gas lights for emergency use. Increased reliability of the electric system has now made this precaution unnecessary. Light failures on well-maintained electric service now average in the neighborhood of 1 per 1,000,000 miles.

CANDLE PERIOD 1825 TO 1850

3 The railroad companies first developed the candle light by placing the candles in proper sockets, putting chimneys around them and deflectors on top of the chimneys. Later the candle flame was kept at a predetermined focus by a spring in the candle socket. The candles used in this country were made from the solid distillates of petroleum formed in molds under pressure, while in Europe wax and tallow dips were chiefly used.

OIL PERIOD 1850 TO 1875

4 The railroad companies recognizing the need of better illumination tried various forms of oil lamps, and in this development the lamps were always the latest improvement as regards safety and reliability. The oils used in this country were distillates of petroleum differing in flame and fire test in the various states and on different roads. The oils used in Europe were vegetable oils, because of the high cost of importing mineral oils and because of the supposed danger from fire. The danger of fire from oil of either class depended, however, more upon the method of containing than in the kindling point, except in the very low flash point distillates of petroleum which were not used directly in railroad work.

5 The European practice for oil lamps was to use rape seed and Colza vegetable oils which are irregular in composition and congeal in cold weather. The difficulties met with in the burning of this oil in Europe led to the development of central draft burners, which were also used in this country to burn heavy oils. The early types of mineral oil burners were merely reservoirs into which fibrous wicks were dipped through suitable holders. Two wicks feeding one flame was the next development, allowing a more copious supply of oil to the flame and a brighter light resulted.

6 The Argand and Belgian burners are representative types of the central draft burner. The Argand burner has a tube and annular wick providing a strong draft of air both to the inside and to the outside of the flame. This results in a high temperature and a bright white light. This burner was the forerunner of several central draft burners used for car, locomotive and station lighting, where a high candle power was desired. A maximum of 60 c.p. has been obtained from one lamp of a size suitable for railway cars by this central draft method. The Belgian lamp represents the highest improvement in delivering a current of air both inside and outside the flame so that it will impinge on the flame at the proper point to produce rapid combustion. Other burners of this type are the Acme, the Student Lamp, and the Westlake lamp.

7 The reservoirs of these lamps were generally of metal and either double glass or specially tough single glass chimneys were used. When the lamps were placed in the center of the ceiling of the car, smoke bells were provided to take off the products of combustion and to protect the flame from drafts. The oil lamps had been generally superseded for first-class service by other illuminants before the development of the incandescent mantle, and therefore, has not been used with the oil lamp in train lighting.

GAS PERIOD 1875 TO 1900

8 About 1870 coal gas was introduced as a head-end system on several railroads as a means of lighting cars. A reservoir in one of the cars, known as the guards van, was filled with gas secured from the city mains and then slightly compressed by water displacement. The reservoir which had a capacity of about 100 cu. ft. was generally square in shape and was made of strong canvas reinforced by wood ribs. Gas was conveyed to the lamps, which burned about 5 cu. ft. each per hour, through iron pipes with rubber hose connections between cars.

9 Nearly ten years later Mr. George Westinghouse devised the "air gas" or carbureted air system. A heavy cylinder, suspended underneath each car, was filled with such material as sawdust, felt, etc., saturated with some light hydrocarbon such as benzine. Compressed air was passed through the tank and, becoming saturated with the hydrocarbon vapor, was conveyed to the lamps where it was burned. The system was never ad-

vanced much beyond the experimental stage, as changes in temperature affected the carburation to a great extent.

10 Carbureted gasolene, made by mixing air and gasolene in suitable proportions was the next system to be used. The best example is probably the Frost Dry Carbureter System. The essentials of this system are that the gasolene be kept from leaking and that the gas be burned as soon after generation as possible without undergoing any change of temperature. The Frost system accomplishes this by locating on the roof of the car directly over each lamp, the carbureter which consists of a chamber filled with wicking soaked in gasolene through which air from the air-brake system is passed at a low pressure, the gasolene vapor being supplied to a burner directly underneath. The products of combustion are passed out through a smoke bell which leads around the carbureter, thus tending to keep the gas at the proper temperature in all weather. The difficulty of maintaining this proper temperature, however, was the chief drawback of the system.

11 The extensive manufacture of calcium carbide by the electrochemical process gave the acetylene system a great impetus. Examples of the acetylene system in extensive use were first, the Adlake system in which about 50 lb. of calcium carbide, placed in a portable steel container installed in a special panel in the car, is arranged so that gas is generated automatically as required by water dropping upon the carbide. In order to obtain this automatic supply, a small tank is placed underneath the car and connected to the steel container, which automatically produces the gas because of the back pressure of the water supply when a certain amount of gas has been generated.

12 The Avery system resembles the Adlake system, except that the generator is placed underneath the car body.

13 The third system used to some extent, is known as the Commercial, and in this acetone, a liquid distilled from wood alcohol and having the property of absorbing acetylene under pressure, is used. The gas is stored in tanks filled with asbestos disks saturated with acetone, and is given off as the pressure in the tank is reduced.

14 Acetylene gas gives a more intense light than any of the other oil or gas systems, but it is explosive and this fact has militated against its extensive use.

15 In 1867 Pintsch, in Berlin, devised the system of gas illumination which bears his name. The distinctive features of this system are, the manufacture of gas from crude petroleum by heat distillation, which gas is not decomposed by compression and subsequent expansion and a regulator which controls the supply of gas to the lamps at a very low pressure for atmospheric burning. Because of its reliability and increased storage capacity, obtained by charging the reservoirs at a high pressure, the Pintsch gas is very extensively used for car lighting. Other systems previously tried, namely, coal, water, and natural gas, did not have these distinctive features and up to that time, were not a success. The substitution of mantle burners for the ordinary fish-tail burners has greatly improved the quality of light, and enabled the Pintsch system to compete on more equal terms with electric lighting.

16. The Pintsch system requires a separate plant with appliances for generating, compressing and storing the gas, also suitable tank cars and pipe lines for conveying the gas to the car reservoirs.

ELECTRIC PERIOD 1900 TO DATE

17 Early in 1881 the London, Brighton and South Coast railway had in operation a straight storage battery system with a battery under each car, the battery being charged during the layover period at terminals.

18 Bichromate primary batteries were tried on trains in France during the year 1885. The scheme was abandoned, however, when the first set of batteries had run down and renewal of the electrodes and acid was necessary.

19 About this time both the Pennsylvania Railroad and the Boston and Albany tried out the straight storage system. The general opinion was that the system was practical but too expensive for general use.

20 In 1888 the Chicago, Milwaukee and St. Paul had in service a train equipped with a head-end system consisting of a boiler, engine, generator, etc., installed in a separate car for supplying light to the train. The present system, developed later, uses steam from the locomotive.

21 The axle driven generator was developed coincidently with electric lighting. The first installation of this kind consisted of a generator connected to a relatively high capacity

battery and hand-operated main switch and pole changer, regulation being effected through the inherent regulation of the batteries taking the excess current at high speeds, thereby holding the voltage nearly constant by armature reaction. Later the brushes were automatically shifted with the speed for voltage regulation. The next step was to allow the belt to slip at speeds in excess of the speed required to deliver current at normal voltage.

22 Various early and intermediate systems were proposed, among which may be mentioned:

a A head-end system deriving power from a fan-driven dynamo on the pilot.

b A track wire system, especially through tunnels, for lighting cars from a trolley.

c A track wire system for charging batteries from an auxiliary trolley while cars were passing through electrified territory.

23 The working out of the early tried system resulted in the more or less general adoption of the following: (*a*) the straight storage system, (*b*) the head-end system, (*c*) the axle generator system.

STRAIGHT STORAGE SYSTEM

24 In this system, each car is provided with a storage battery, which must be charged at terminals during the layover period. A consideration of the requirements for successful operation reveals the following essentials:

The capacity of the battery must be in excess of the demand for current to operate lamps, fans, etc., for the longest run between charging periods.

The power plant, or other outside source of power, must be of sufficient capacity to meet the maximum demand for charging current.

The layover time at terminals must be sufficient to cover all necessary shifting and charging of the batteries at the proper rate.

The yard must have a sufficient number of tracks provided with charging outlets, so arranged that the charging of batteries will not interfere with shifting operations.

25 Two considerations enter into the determination of the

size of the battery; the demand for current and the weight of the battery. The demand for current will vary with the type of car used, the number and size of lamps and other electrical apparatus installed and the time during which this apparatus is used. The battery must be divided into units of a weight that may be readily handled.

26 The batteries in general use in train lighting service have a rated capacity of approximately 300 ampere-hours. This is about the maximum limit of capacity for Plante's type batteries having weight low enough for convenience in handling.

27 The great majority of cars on which this system is used are equipped with a 64 volt lead-lead, or nickel-iron battery with cells connected in series. A few cars are operated at other voltages, viz. 26, 30, 32 and 110. It was the practice of one railroad some years ago to operate at 25 volts, using 12 lead-lead cells in series, with two or more 12 cell sets on a car connected in multiple.

28 Two battery boxes are generally provided and secured to the under side of the car, one on each side equidistant from the ends of the car, and with the front or door side slightly back of the line of outside finish. The cells are put up in double compartment lead-lined wood tanks provided with handles, rollers, etc., for convenience in handling.

29 The two halves of the battery are connected in series and leads are run to the switchboard in the end of the car. Taps are taken off these leads at the battery terminals and run to charging receptacles, conveniently located on each side of the car.

30 The ampere-hour meter is coming into general favor as an indicator of the state of charge of the battery. As an adjunct a shunt trip circuit breaker is sometimes installed, the connections being such that, when the battery is fully charged, the pointer on the meter closes a circuit which energizes the shunt trip and opens the breaker, thus cutting the battery off charge.

31 The source of power available for charging purposes at terminal yards must be of the proper voltage and of capacity sufficient to meet the maximum demand. The charging voltage provided is usually 50 per cent higher than the normal voltage of the battery. Hand or automatically operated resistance devices are provided for reducing the voltage of the individual charging lines to the proper point.

32 The batteries are charged while on the car under normal conditions. When the layover period is short it is sometimes necessary to exchange a discharged battery for one fully charged.

33 The operating schedule of cars equipped with this system must be worked out to allow sufficient layover at terminals to permit the charging of batteries and shifting of cars. Anything that restricts the shifting operations necessitates either additional yard trackage, additional motive power, or both. As cars produce a revenue only when in service, it is the aim of the transportation department to decrease the layover period to a minimum.

HEAD-END SYSTEM

34 The head-end system consists essentially of a steam-driven generator located in the baggage car or on the locomotive. Proper controlling apparatus is provided and train lines are run from the generator through the entire length of the train, flexible connections being used between cars.

35 So far as can be learned the first head-end system was installed by the Pullman Palace Car Company in January 1887, in some of their composite cars. A three-cylinder Brotherhood engine, direct-connected to an Eickemeyer generator, rated at 80 amperes and 80 volts full load at 1200 r.p.m. was used. These cars were operated in the Atlantic Coast Line Special running between Jersey City, N. J., and St. Augustine, Fla. In June of the same year the Pennsylvania Limited was equipped with the same type of apparatus. In both of these trains the individual cars were equipped with batteries.

36 This system of lighting was afterwards taken up by other eastern railroads but at the present time it is used chiefly on roads operating west of Chicago. Following the Brotherhood engine, the Westinghouse standard engine, the De Laval turbine and the Curtis turbine entered the field in turn.

37 The Pennsylvania Railroad conducted a series of experiments with the "Gould Booster" head-end system, which consisted of a compound-wound turbine-driven generator mounted on the locomotive, with a motor-generator booster and automatic switches mounted on the tender. Four train line wires were run underneath the cars and were joined together by connectors that uncoupled automatically when the cars were parted. A

battery and transfer switch were provided on each car. In this system, unlike any other in use at that time, the entire control was automatic after the generator was started.

38 The head-end system of today comprises the following apparatus:

A generator, usually steam turbine-driven, placed in the baggage car or on the locomotive, and furnished with steam from the locomotive.

The necessary indicating, regulating, and controlling apparatus placed near the generator and in an accessible position.

Train line wires of the proper size on each car and running the entire length of the train, flexible connections being made between cars, in the vestibule.

Batteries, consisting of a suitable number of cells connected in series and placed in battery boxes attached to the under side of the cars.

Lamp regulators are sometimes installed in the cars to compensate for the line drop and to maintain constant voltage at the lamps.

39 The successful operation of this system requires that a sufficient amount of steam at the proper pressure be provided when lighting is necessary. As it is the object of the transportation department to get trains to their destination on time, lack of steam is felt first by the lighting system, the pressure being reduced or steam cut off entirely so that the schedule may be maintained.

40 When the train is broken up en route, it is obvious that each section must either be equipped with a battery to insure light until the train is again made up, or provided with some auxiliary light.

41 A member of the train crew must be capable of operating the generating apparatus and of making running repairs and adjustments en route.

42 Head-end systems are generally operated at 64 or 110 volts, although in late years the introduction of tungsten lamps has gone a long way towards eliminating the need of the high voltage equipments and comparatively few railroads are now using 110 volts.

43 It is apparent that the use of the head-end system must be restricted to a few trains having assigned runs or that

it must be extended to cover all the cars operated by the railroad in electrically-lighted trains.

AXLE GENERATOR SYSTEM

44 The axle generator systems used in this country comprise the following principal parts:

a An axle-driven generator mounted on the car truck. (Abroad where rigid trucks are used the axle generator is frequently secured to the under side of the car body.)

b A suspension by which the axle generator is supported from the truck frame.

c A drive, connecting the armature shaft to the axle.

d A regulator for controlling the voltage and output of the generator at all train speeds.

e An automatic switch designed to open on reverse current for the purpose of preventing discharge of the battery through the generator.

f A regulator for controlling the voltage impressed on the lamp circuits.

g A battery of a suitable number of cells to supply current when generator current is not available.

45 For the successful operation of the system, the following requirements must be met:

The polarity of the generator terminals must remain unchanged with a movement of the car in either direction.

At all train speeds, from the cutting-in speed of the generator to the maximum, the generator output and voltage must be maintained within the desired working limits.

The generator must be automatically connected and disconnected from the battery circuit as the train speed rises above or falls below the critical speed.

The lights may be burned at any time and the transfer of this load from the battery to the generator and vice-versa must result in no appreciable change in the candle power of the lamps.

The voltage impressed on the lamp circuit must be maintained within such limits as will give satisfactory illumination and reasonable life of lamps.

46 The early axle generators were practically all of the constant current type. The generator maintained a constant current, only that portion going to the battery that was not required by the lights. The generator output could be varied by hand adjustment, and it was necessary to adjust the generator output so that there would be an approximate balance between the battery input and battery output, account being taken of the battery ampere-hour efficiency. This requires consideration of the following factors: (*a*) the ampere capacity of the generator, (*b*) the ampere-hour demand during the trip. The demand varies with the number and size of lamps installed, the run in which the car is operated, the number and duration of stops, and the proportion of total running time of the train below cutting-in speed. A change of the car from local to through service, or vice-versa, required a complete readjustment of the regulation. A failure so to readjust would quickly result in battery deterioration, the plates becoming sulphated with a low generator output, while a corresponding high output would boil away the electrolyte. In a variation of this method the battery was charged at a definite rate, generally the normal charging rate, while the generator output varied with the lamp load. In more recent systems constant current is maintained until the generator voltage, increasing as the counter e.m.f. of the battery, reached a definite value, after which one of two results would obtain, depending upon the design of the control apparatus. In one the generator voltage was held approximately constant producing what was known as the taper charge or a gradual decrease in the rate of charge. In the other, the generator voltage was automatically reduced to the "floating" e.m.f. of the battery and maintained approximately constant. This was known as "stop charge" regulation.

47 Both methods originally depended upon the operation of a voltage coil actuating a switch, which in turn changed the field current by means of resistance. Premature closing of the battery stop charge switch resulted in undercharging and consequent sulphating, while delayed closing caused overcharging and the boiling away of the electrolyte.

48 These devices are being superseded by a system using a voltage coil in the regulator instead of the voltage switches previously mentioned. Here the current output is maintained constant to the full generator capacity

regardless of the current demand, and the battery must supply all current in excess of the capacity of the generator. As the battery becomes charged the generator voltage gradually increases and at a predetermined voltage the voltage regulating coil takes the control of the generator from the current regulating coil and when the battery becomes fully charged the current decreases until the battery is "floating on the line." As the voltage on the battery on charge is approximately 30 per cent higher than on discharge, it is necessary to provide some means of lamp regulation in order to keep a constant voltage on the lamps. In the early axle generating systems this was accomplished by introducing a fixed metallic resistance into the lamp circuit at the instant the automatic switch closed, transferring lamp load from battery to generator. This method, however, was not altogether reliable as the generator voltage varied with the condition of the battery and the number of lamps burning. The variation of the voltage due to these conditions soon led to the development of a variable metallic resistance in place of the "fixed" resistance. In this method of regulation the amount of resistance in the circuit was varied by means of the operation of a small auxiliary motor operated by a voltage relay or other means.

49 A later type of lamp regulator employed, as a means of regulating resistance, a series of carbon blocks, the resistance being varied by varying the pressure on these blocks, the variation of pressure being determined by a pilot voltage coil connected across the lamp mains.

50 In this country it is the general practice to support the axle generator from the truck frame. When first applied, the generator was placed between the axle and the truck-end sill, this arrangement being known as "inside suspension." The generator was not easily accessible for inspection and repairs, and at the present time it is placed outside of the truck frame, this arrangement being known as "outside suspension." There are four general methods of carrying the generator from the suspension framing, viz. bottom pivoted, top pivoted, parallel link and sliding. The bottom pivoted was first used but at the present time the parallel link suspension is in more general use.

51 The drive generally used employed a rubber-filled canvas belt running on pulleys on the axle and the armature shaft.

The axle pulley as first used was cast iron mounted directly on the car axle, the bore of the pulley conforming to the taper of the axle, but on account of inequalities in the axle which was hammered or rough-turned, it was wrapped with tarred paper. The axle pulley at present in use is of pressed steel, mounted on a steel bushing, the bushing being secured independently to a turned seat on the axle, and the pulley mounted thereon. Belt tension is provided by means of springs which also afford relief to the belt due to the movement of the car axle with respect to the truck frame. One spring is generally used when the generator has top, bottom or sliding suspension and two springs with the parallel link suspension. Chains of the silent type have also been tried and have the advantage of positive action and decrease in bearing pressure, but the wear of the links both on the face and the pivot sprockets has been excessive. Belts of V-section have been tried and would seem to have the same advantages as the chains, but it is found that in winter the bottom of the V-groove in the sheaves packs with ice and snow, and driving power is lost. Neither the chain nor V-belt requires tension device.

52 A form of shaft drive, from a bevel gear on the car axle, through an extensible shaft with universal couplings to a generator carried from the car body, is being tried. A gear drive, with the generator mounted on the car axle after the manner of the street car motor mounting has also been prepared and will soon be tried. The latter will no doubt require a track pit for the inspection and repair of the generator.

53 Plain bearings with ring oilers were used almost exclusively until five years ago. With this method of lubrication, it was necessary to carry the oil level so high that it frequently entered the generator frame and damaged the armature and field coils. To overcome this trouble, a form of wick oiler was tried which, however, proved unsatisfactory. The next improvement was a combination ring and chain oiler, which is now in general use on one type of machine. Waste-packed bearings have also been extensively used and have given no trouble on account of oil entering the generator frame. Considerable trouble, due to hot bearings has been experienced. Ball bearings for axle generators were introduced in England about 1907 and tried in this country in the early part of 1911. Although the use of ball bearings to date has been limited, the indications are that this

bearing will become very popular on future machines. On account of the widely varying temperatures between summer and winter conditions that obtain in the operation of axle generators, it was necessary to develop special oils that would remain fluid at low temperatures. Those now used have a freezing point of 10 deg. fahr. For ball-bearing lubrication grease should be free from acid or alkali; should not oxidize or evaporate; should not gum nor lose its body.

LAMPS AND VOLTAGE

54 The early train lighting lamps were of the ordinary multiple burning type regularly manufactured, and as the demand increased manufacturers took up the development of lamps especially adapted to train lighting. One of the requisites of these lamps is that the smallest size bulb consistent with reasonable life of lamp be used.

55 Among the early lamps was a foreign-made carbon-filament lamp operating at 4 watts per c.p. Some time later, a carbon lamp operating at 3.5 watts per c.p. was developed by American manufacturers.

56 It was found that the future of electric car lighting depended to a large extent upon whether or not lamps of still higher efficiencies could be manufactured in large quantities. About this time the tungsten lamp was introduced in regular multiple burning service and investigations were immediately started with a view to adapting it to car lighting. The use of tungsten lamps, however, was looked upon with some doubt on account of the extremely fragile nature of the filament, especially in the smaller sizes. Eventually a 25-watt lamp was developed. This did not improve conditions appreciably, since many of the carbon-filament lamps were approximately of the same wattage. Further investigations resulted in the development of a 15-watt lamp which improved conditions sufficiently to warrant its use in large quantities and made it possible to light a car at a reasonable current demand. With the introduction of tungsten lamps the standard voltage of axle generators has been changed from 60 to 30 volts.

57 Coincident with the introduction of tungsten lamps into train lighting service there was introduced the so-called "hot filament" system. The basis for this system was the belief that a tungsten filament of the early type when heated to about 400

deg. fahr., was less fragile than when cold, so that by maintaining the filaments at this temperature the life of lamps could be materially increased. A circuit switching arrangement was provided whereby the lamps were operated across 30 cells of battery when light was desired and across the two remaining cells of the battery at other times, this being sufficient to keep the filaments at a dull red. Later, the introduction of the drawn wire filament rendered this system unnecessary, and it has been abandoned.

58 Metallized filament lamps with small opal bulbs, operating at about 2½ watts per c.p. were developed for berth lighting. These lamps gave satisfaction at first, principally because berth lights had never before been furnished. As the public became educated to their use, the demand for more light in berths became pronounced, so that it has been found necessary to use the 15-watt tungsten lamp as a berth lamp, and the use of metallized filament berth lamps is decreasing very rapidly.

59. Tungsten lamps for train lighting purposes are now furnished in 10, 15, 20, 25 and 50-watt sizes, although the bulk of demand is concentrated upon 15 and 50-watt lamps. Spherical bulbs are used almost exclusively, 3¾ in. in diameter for the 50-watt lamp and 2 5/16 in. in diameter for the smaller sizes.

60 The voltage of the lamp is necessarily dependent upon the voltage of the generating system. An investigation of the records of the principal railroads shows the use of the following voltages at the present time: 24, 26, 30, 32, 60, 64 and 110. The bulk of the demand for train lighting lamps is concentrated on two voltage groups, the 30 volt, which includes lamps between 25 and 34, inclusive, and the 60 volt, which includes lamps of from 50 to 65 volts, inclusive.

61 From an engineering standpoint, it is highly desirable that as many of the voltages, wattages and sizes of bulb as is possible be eliminated. From a manufacturing standpoint, it is desirable that some variation be allowed in the voltages, as a large number of high-voltage and low-voltage lamps are accumulated in the manufacture of lamps of a fixed voltage. This brings up the question of the inspection of lamps before purchase.

62 On account of the method of manufacture of train lighting lamps, manufacturers have found it most convenient to inspect the lamps for initial variation by holding the lamps at a fixed candle power, allowing the voltage and current to vary.

TABLE 1 METHODS AND DISTRIBUTION OF CAR-LIGHTING SYSTEMS

Railroads	Cars Lighted by Electricity	Cars Lighted by Other Means	Straight Storage		Cars with Turbine	Head-End System		Axle Generator System		Owned and Operated by Railroad Company	Owned and Operated by Pullman Company	Cars Contracted for	Number Cells Lead Battery	Number Cells Nickel, Iron, Alkaline Battery
			No.	Volts		No.	Volts	No.	Volts					
A., T. & S. F.	682	1120	0	0	0	0	0	682	30	632	50	50	11552	25
B. & A.	7	434	0	0	0	0	0	0	0	7	0	0	708	0
B. & O.	246	1105	28	60	11	67	64	19	60	143	103	0	3028	100
	..	..	..	..	..	..	..	29	30	..	..	..	..	..
Can. Pac.	165	2273	0	0	0	0	0	128	24	128	0	37	3072	220
C., St. P., M. & O.	58	265	0	0	4	0	64	2	60	2	0	0	192	0
C. of Ga.	6	258	0	0	0	0	0	6	30	0	0	0	80	50
C. & A.	103	100	56	32/64	0	0	0	34	32/64	103	0	0	0	0
C. & E. I.	93	107	2	32	0	0	0	77	32	79	14	0	1264	0
C., B. & Q.	710	591	0	0	52 cr	52	60	2	30	0	0	0	5508	0
	..	..	..	..	16 en	16	64	..	..	..	..	..	..	..
C., M. & St. P.	990	368	30	64	58	58	64	9	64	930	..	60–64	3872	0
	..	..	..	..	..	..	..	2	32	..	..	..	..	..
C. G. W.	70	102	0	0	0	0	0	70	30	70	0	0	736	216
C. & N. W.	672	1000	0	0	23	200	64	17	60	572	100	0	2200	25
	..	..	..	..	..	..	..	35	30	..	..	..	..	..
C., R. I. & P.	344	805	0	0	0	0	0	301	30	0	0	36–30	5024	125
	..	..	..	..	..	..	..	6	60	..	..	1–60	..	..
C., C., C. & St. L.	46	552	2	0	0	0	0	44	32/64	103	0	0	0	0
D., L. & W.	44	738	0	0	0		0	44	30	0	30	0	704	0
	..	..	..	..	..	.		1	60	..	..	..	..	..
D. & R. G.	30	348	0	0	0	0	0	29	32	0	0	0	512	0
E. P. & S.W.	17	46	0	0	0	0	0	17	30	0	0	0	272	0
Erie.	156	1788	74	30	0	0	0	72	30	146	0	10–30	2448	0
G. N.	551	402	11	110	45	550	110	1	30	953	0	0	3040	0
G. R. & I.	33	86	28	60	0	0	0	1	60	33	0	0	912	0
	..	..	..	..	..	..	..	4	30	..	..	..	..	..
G. T.	106	0	0	0	0	0	0	15	24	31	75	0	840	0
	..	..	..	..	..	..	..	15	30	..	..	..	..	..
	..	..	..	..	..	..	..	1	60	..	..	..	..	..
F. W. & D. C.	22	31	0	0	0	0	0	22	32	0	0	0	336	25
Ill. Cen.	330	770	10	30	0	0	0	320	30	240	90	0	2240	3050
	..	..	..	..	..	..	..	3	60	..	..	..	..	..
K. C. So.	21	57	0	0	0	0	0	18	30	0	0	0	384	0
L. V.	219	340	0	0	0	0	0	196	32	0	0	23–30	3480	24
L. I.	113	404	3	64	0	0	0	109	32	0	0	0	3680	125
L. S. & M. S.	139	634	0	0	0	0	0	72	30	0	0	0	0	0
	..	..	1	64	..	..	..	47	60	..	..	..	3312	..
L. & N.	67	558	1	32	0	0	0	59	32	0	0	532	1120	75
M. P.	208	739	1	32	0	0	0	109	32	0	178	0	1824	0
M. & O.	6	119	0	0	0	0	0	6	..	6	0	0	0	0

TABLE 1—*Continued*

Railroads	Cars Lighted by Electricity	Cars Lighted by Other Means	Straight Storage		Cars with Turbine	Head-End System		Axle Generator System		Owned and Operated by Railroad Company	Owned and Operated by Pullman Company	Cars Contracted for	Number Cells Lead Battery	Number Cells Nickel, Iron, Alkaline Battery
			No.	Volts		No.	Volts	No.	Volts					
M. & St. L.....	19	76	0	0	0	0	0	19	32	17	2	0	304	0
M. C.....	60	438	0	0	0	0	0	18	60	0	0	12–30	1248	0
	..	..	..	..	..	..	..	30	30	..	..	..	..	..
M., St. P. & S. St. M....	65	328	0	0	0	0	0	65	32	0	20	0	1072	0
N.Y.C.&H.R.	376	528	0	0	0	0	0	376	32-64	376	0	48	6624	54
N.Y.C.&St.L.	2	..	0	0	0	0	0	2	60	0	0	0	64	0
N.Y., N. H. & H.....	330	2103	0	0	0	3	110	113	30	330	0	36–60	7280	270
	..	..	..	..	..	..	..	178	60	..	..	..	..	..
N. P.....	667	492	0	0	53	658	64	9	64	0	0	0	28621	0
N. R. of Mex.	36	..	0	0	0	0	0	31	32	9	22	5	592	0
N. & W.....	77	382	0	0	0	0	0	13	30	13	30	64	0	1600
O. S. L.....	84	198	1	32	6	82	64	1	32	282	82	23	480	0
O. R. & N....	56	55	0	0	7	55	64	1	60	131	0	0	436	0
Pullman.....	2400	1864	6	0	0	0	0	2400	30	0	0	0	38400	0
P. & L. E.....	4	134	1	60	0	0	0	2	30	0	0	0	96	0
	..	..	1	30	..	..	..	..	..	..	..	..	..	..
P. M.....	102	259	0	0	0	0	0	102	32	102	0	0	1632	0
St. L. & S. F..	246	388	0	0	0	0	0	236	32	0	76	10	3820	0
St. L. & S. W.	15	208	0	0	0	0	0	4	60	0	11	0	304	0
	..	..	..	..	..	..	..	11	30	..	..	..	..	..
So.....	258	..	0	0	0	0	0	228	30	0	0	30	4128	0
S. P.....	145	1400	2	0	0	71	64	72	32/64	145	0	0	2200	0
S. P., L. A. & S. L.....	76	97	1	0	0	55	64	20	32/60	24	0	0	608	0
T. & N. O....	76	533	0	0	0	0	0	16	64	16	47	0	512	0
T. & B. V.....	16	12	0	0	0	0	0	16	32	0	0	0	176	125
U. P.....	221	346	0	0	32eng	202	63	13	64	221	0	44	2020	64
	..	..	..	..	..	..	..	6	32	..	..	..	..	..
W. & L. E....	4	72	3	0	0	0	0	1	..	4	0	0	0	0
W. P.....	86	..	0	0	0	0	0	85	32	0	0	0	0	0
	..	..	..	..	..	..	..	1	64	87	..	..	1392	..
Penna. East..	1092	2310	845	60	0	0	0	19	60	1226	0	489	53632	2550
	..	..	..	..	..	..	..	228	30	..	..	..	..	..
Penna. West.	581	717	526	60	0	0	0	5	30	581	0	54	22592	50
	..	..	..	..	..	..	..	50	60	..	..	..	..	..
Vandalia.....	53	116	42	60	0	0	0	11	60	53	0	0	2128	0
Wabash.....	..	..	..	..	..	..	..	..	..	..	..	..	..	..
Total, June 30, 1912.	13786	29275	1680	..	307	2069	..	7060	..	7847	1330	1849	242932	3817
Total, June 30, 1911.	11017	33634	1372	..	192	3185	..	5900	..	..	..	..	202744	..

From the railroad standpoint, this method of inspection seems to be wrong, as the voltage of the circuit is maintained at a very closely fixed value in actual service. If the manufacturers could be persuaded to do so, it would be much more desirable to have lamps inspected by holding them at a predetermined voltage, allowing the variations to take place in the wattage and candle power.

63 According to the present method of inspection, a variation of 5 per cent above and below in voltage and 8 per cent above in watts is allowed. If the demand is concentrated too greatly upon one type of lamp, it will probably be necessary to increase these limits. On the other hand, if the railroads among themselves can standardize their voltages close enough for each voltage class to permit the use of any of the lamps on any railroad, the demand for any one voltage lamp will probably not be great enough to make the increasing of the limits of inspection above mentioned necessary.

64 Until recently little attention has been given to the development of reflectors adapted particularly to railroad service. There are several very efficient reflectors now on the market which in some cases have increased the intensity of the light on the reading plane in a passenger car as much as 65 per cent without changing the current consumption or the candlepower of the lamps. To secure the best results, attention must be given to the proper location of the filament relative to the reflector.

65 The question of sockets also plays a very important part in car lighting. The socket must be sufficiently rugged in its construction to withstand the jar and vibration encountered in the handling of passenger equipment cars. It must also be so constructed that when the lamp is screwed home there will be sufficient spring in the inner shell and contact to lock the lamp in position, holding it so tight that it will not become unscrewed by the jar and vibration.

STANDARDIZATION

66 The Association of Railway Electrical Engineers has made recommendations as to the standardization of many of the details pertaining to electric car lighting and a number of the railroads of the country have already indicated their intention of following these recommendations by changing their equipment to conform to the recommended practice.

67 The standard voltages recommended by the Association of Railway Electrical Engineers are:

60 volts (nominal) for straight storage and head-end systems

30 volts (nominal) for axle generator systems

68 The lead battery has been fairly well standardized and the construction recommended includes a two-compartment lead-lined tank with rubber jars. The principal variations from the standard in the lead batteries are that some roads are now using lead covers in place of the rubber covers, this change being made to lessen the danger to battery repairmen on account of gas explosions, and the difference in the method of making the connections to the battery posts.

69 The differences in batteries are not of such a character as to preclude the interchange of batteries on different roads.

70 The nickel-iron battery is manufactured only by one company and therefore the battery is more easily obtained to standard construction and dimensions.

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DISCUSSION

R. M. Dixon. Referring to Pars. 9 and 10 relative to gasolene systems, by the use of mantles and improvements in apparatus both of these methods are very acceptable for car lighting, especially for cars in service on branch lines or on roads where Pintsch gas is not available. Mantles permit of the use of a blue gas which is non-illuminating of itself. Gas of such a quality can be produced at temperatures below that of zero degrees fahrenheit. If it be borne in mind for a moment that the characteristic controlling the absorption of vapors is independent of pressure, but dependent upon volume, it will be readily understood that air can be carburetted to an equal amount whether at a pressure approximately atmospheric, or at pressures greater than atmospheric, the illuminating value is proportionately less. Therefore, if air be supplied to the carbureter under pressures which are increased to harmonize with increased temperatures and, consequently increased vapor tension, the expanded gas will be of uniform quality. It is, therefore, only necessary to control the supply of air to the carbureter with a proper thermostatic factor controlling the pressure in the carbureter and the gas, when afterwards expanded to the proper pressure for burning, will be of a uniform, constant and suitable quality. This is accomplished automatically. Such a system is very reliable and is operated at a cost of approximately ½ cent an hour for 100 candles light.

The statement that "Pintsch gas is very extensively used for car lighting," hardly conveys an adequate idea of the very general and world-wide use of the Pintsch system. There is not a country on earth where it is not used; there is no single appliance used in passenger car construction or equipment that has had the wide and general use of Pintsch gas. The total number of cars lighted by this system aggregates five times the entire passenger equipment of the railroads of the United States. It is a most economical system, costing on the average about the same as air gas, namely, about ½ cent per hour for 100 candles light. Approximately 12,000 cars per annum are now being equipped with this system, this number being equivalent to about 25 per cent of the entire passenger equipment of the United States.

As regards electric light, axle-driven generators provide the most flexible and at the same time the most economical equipments in operation and while it has heretofore been necessary to provide the various apparatus for each car, as listed in Par. 44, a

recent development, already in considerable use in Europe, is now being demonstrated in the United States. This system embodies a self-regulating dynamo without a special regulator for controlling the voltage and output of the generator at all train speeds and without a regulator for controlling the voltage imposed on the lamps. A few of these equipments operating on suburban service cars, give sufficient current to maintain the lights and battery charges and the lamp voltage does not vary ½ volt either above or below normal.

W. L. Bliss.[1] The straight storage system appeals to the railroad man because of its simplicity. This simplicity is, however, more apparent than real. When it is necessary to light a number of cars, the question of terminal facilities for taking care of batteries and properly charging them becomes of great moment, and it is doubtful if any of the large trunk lines in the East can economically operate all of their cars with this system. The great objection to the straight storage system when operated on a large scale is the fact that so many cars must be tied up at terminals while their batteries are being charged. It has never been customary to equip straight storage cars with any regulating apparatus of an automatic character, although at the present time there is a growing tendency to adopt the ampere-hour meter with the automatic cut-off device on straight storage cars. There is no question, however, in the writer's mind but that the use of a lamp regulator would prove a source of economy in lamp renewals on straight storage cars.

Reference was made in this paper to the possibilities of axle head-end lighting, a subject somewhat overlooked by railroad companies and manufacturers. In many instances the head-end axle system is the most economical arrangement that can be devised. The company with which the writer is connected installed such a system about six years ago on the Baltimore and Ohio Railroad. A generator large enough to light four cars was installed on the truck of a combined baggage and smoking car, and all cars in the train were equipped with lamp regulators and standard train lines and batteries. The combined car carried the regulating apparatus for the generator, together with its battery and lamp regulator. This system remained in operation for a number of years, but it had the disadvantage that it was neces-

[1] Chief Engineer, United States Light & Heating Co., Niagara Falls, N. Y.

sary to keep the cars equipped with batteries in the same train with the car equipped with the generator.

About nine years ago the writer installed on the Pennsylvania Railroad a head-end axle system, which was novel in many respects. The generator was mounted on the truck of the locomotive tender. It was of unusual capacity, having an output of about 350 amperes at 80 volts, and was constructed along the lines of a street railway motor, being journaled upon and geared to the tender axle. This generator was capable of furnishing current for the longest trains of the Pennsylvania Limited. The objection to this system was that it was impossible to keep the train made up of cars equipped to operate with the tender generator. The introduction into the train of any cars not so equipped broke up the operative combination.

There are three general methods of lighting railroad cars by the head-end system: (*a*) the customary steam-driven generator in the baggage car; (*b*) the steam-driven generator on the locomotive; (*c*) the axle-driven generator mounted upon the truck of the locomotive tender. Such a machine has many advantages: it can be made large; it requires no pole changer; it can be gear driven, thus eliminating the uncertainty of a belt. The tender-generator arrangement has all the advantages of the steam head-end systems and few of their disadvantages. Its possibilities have by no means been exhausted.

Coming now to the subject of axle lighting of individual cars, this system has apparently come to stay and seems to possess all the desirable features of flexibility and independence: it is self-recuperating; a car could be started out on the road with the storage batteries in a totally discharged condition, and within perhaps a few hours, or at most a day or two, the lighting would be in first-class operative condition, without anybody having expended any effort or time to bring about such results.

To within the last five or six years the axle generator was so difficult to maintain and was out of order so much of the time, that nobody gave much thought to other parts of the system, or devoted any great amount of time or effort to their refinement. Today, however, since the axle generator has been so fully developed, problems connected with economical operation are being taken up. Formerly no difficulty was experienced with overcharged batteries; the whole problem was to keep the batteries from being in a state of total discharge. The cry now seems to

be that batteries are being overcharged, with consequent disintegration. This is the best evidence that the axle generator is keeping up to the game. Apparently it is generating more electrical energy than is actually required. In some cases large generators and large batteries may be required, due to slow schedule and long lay-overs. To insure that the batteries will always be charged, the generator must have a large margin of capacity, and unless means are provided to prevent it, overcharging of the batteries will result. This has led inventors during the last few years to devote a great deal of attention to perfecting regulating apparatus.

Ampere-hour meters have been made with automatic cut-off attachments so that when a certain predetermined charge has been sent into the battery, the generator can be made more or less inoperative and the charging discontinued. It is a fair assumption that when the voltage in a 30-volt system has reached 42 or 43, the batteries are fully charged. The apparatus for determining this point is arranged so that it puts into action certain instrumentalities which reduce the effectiveness of the generator and stop, or taper, the charging current flowing to the battery; hence the expressions, "stop-charge" and "taper-charge" devices. All such devices operate to weaken the field of the generator, which weakening is usually superimposed upon, or added to the weakening that is necessary to compensate for changes in car speed.

In addition to performing the function of preventing overcharging of batteries, the so-called stop and taper-charge devices are called upon to protect all of the apparatus in case of certain kinds of accidents, such as the opening of the battery circuit or the failure of the generator current regulator to operate. In fact, recent specifications require extremely accurate generator voltage regulation on the part of these devices that are primarily designed to prevent overcharging.

There are advocates of both the taper and stop-charge principles, but the writer believes that neither of these schemes is quite satisfactory. The ideal device for preventing overcharging would respond at a voltage of approximately 42 or 43 with great accuracy. Immediately after acting at this voltage, the device should become a strict potential regulator, and should maintain the generator voltage at 35, which is the exact voltage at which a fully charged battery floats. With such an arrangement with the battery floating and the generator voltage held at 35 volts,

the generator may pick up any lamp load without materially affecting the floating condition of the battery, which, of course, is zero current. The stop-charge arrangement is unsatisfactory in that its action is abrupt and causes more or less disturbance in the system, and does not leave the generator in condition to carry the lamp load. The taper-charge arrangement is defective in that it allows destructive low-rate currents to flow after the battery has been fully charged. The generator, however, with the taper-charge device is generally in better condition to pick up the lamp load than with the stop-charge arrangement. The arrangement whereby the charging action is stopped at a definite voltage, and immediately thereafter the batteries are caused to float, would seem an almost ideal condition of affairs.

C. D. Young. Where the service can be filled satisfactorily with straight storage lighting, with properly designed storage batteries, and without any lamp regulation, this system is probably the simplest and most economical. When one of the units fails, the entire train is not affected; there is just the one dimly lighted car, whereas the rest of the train is in very fair shape. If the passengers are inclined to read, they can pass from one car to another. I believe that there is a field for the axle generator system of lighting cars where electric service cannot be furnished with the straight storage batteries, as for example, such cars as are operating in the postal car service.

The axle generator system apparently does not require the last three items in Par. 24 in connection with the straight storage system, namely, the power plant, the layover time at terminals to make the necessary shifting and charging of the batteries, and a yard provided with a sufficient number of tracks with charging outlets. I believe, however, that most of the roads operating axle lighting systems find it desirable at times to charge the batteries before the axle lighting car goes out so that they will have the assurance that it will make a satisfactory trip, and in cars where lights have to be used while the car is standing, for instance, when mail is being distributed before the car has left the terminal.

There is one point that has not been brought out by the paper and that is the question of emergency lighting when the system fails. This is rather important in the operating of any cars, but especially in electric lighted cars. Frequently mails cannot be

distributed on account of light failure and a lack of proper emergency light.

The head-end system does not seem to take care of the general service as well as the individual lighting of cars.

In Par. 66 reference is made to the recommendation by the Association of Railway Electrical Engineers as to train lighting. The report to have been complete should have also called attention to the standard adopted by the Master Car Builders' Association as to the fixtures desired in straight storage cars.

J. R. Sloan.[1] Where the storage battery system is applicable, it is believed it can be operated at a lower cost per car month than the head-end system or axle generator system. Unfortunately, the limitations of the storage battery system are such that very few roads in this country can satisfactorily light their cars in this manner, and even on those roads some runs cannot be handled satisfactorily by straight storage methods. From the inherent features of the head-end system it is impossible to select only certain classes of cars to be equipped for electric lighting, since to be operated successfully all of the cars in the train must be wired for the operation of this system, and if many trains are operated it will be found that all classes of cars are included. The head-end system is limited, therefore, to certain trains having assigned cars, or it must be handled as a general proposition and all cars operated over the road must be equipped for this system.

A few roads which had lighted certain trains by the head-end system have abandoned it as the number of cars lighted increased, on account of the difficulties of operation in the transition stage from particular trains with assigned cars to a general proposition where all cars are lighted in this manner. The greater the number of branch lines and the more trains that are broken up at division or junction points to attach or detach cars, the greater these difficulties.

The axle generator system will be more generally used in the future than either the straight storage or head-end system for the reason that each car is an independent unit, and there are practically no limitations with regard to the length of the run or to the time of layover required. The layover of practically all cars in all lines is sufficient to take care of the axle generator equipment under ordinary circumstances.

[1] Engineer, Electric Car Lighting, Penn. R. R., Altoona, Pa.

C. R. GILMAN.[1] The head-end system of lighting was started in the west in the fall of 1888 when the Chicago, Milwaukee & St. Paul Railway Company installed storage batteries on their trains. After two years of operation, owing to inexperience and poor batteries they were discontinued and electric lighted trains depended on oil and gas lamps for an auxiliary. In general this arrangement has continued up to the present time; now, however, it is quite usual to find head-end trains fully equipped with storage batteries as part of the lighting system. The reason for the head-end lighting in western trains is its ease of operating and small investment; also it is very reliable when supplemented with storage batteries. For lighting a solid through train, (say for a 2000-mile run, such as from Chicago to Seattle and return), it cannot be equalled by any other system. Maximum candle power can be maintaiend at all times, and generally batteries can be fully charged during the latter part of the night so that there is no necessity of operating the steam turbine in the day time. Moreover the turbine can often be shut down at night when the train is passing on a heavy up grade so as to give the locomotive all its steam for keeping up running time.

Eighteen solid trains of this type have been in operation for over a year and we have yet to hear of a single failure of electric light. We have had to economize in its use on a few occasions over one or two divisions owing to the failure of a turbine, or parts on the locomotive. Each train averages ten cars and carries three storage batteries of capacity to light all the lamps in the train for three hours, and we have found the capacity sufficient.

The batteries are evenly distributed, one under the baggage car, one in the middle and one on the rear car of the train. The turbine dynamo and the charging of batteries in the trains is done by the baggage man for which he is paid $10 per month. So successful has been this charging of batteries en route that an average of but one battery per month so far is recharged in our terminal yards. The electrical equipment for lighting a ten-car train at 60 volts cost is

Turbine generator and switchboard	$1500.00
Three sets of 32-cell storage batteries	2000.00
	$3500.00

On a question of voltage, 60 is the only practical one for head-

[1] Chief Electrician, C. M. & St. P. Ry. Co., Milwaukee, Wis.

end lighting on account of the ampere load and size of main train wires and connectors. We have about reached the limit in size owing to mechanical reasons, and should a radical change be made on the interior of cars using white ceilings and walls, the lamp load could probably be cut 35 or 40 per cent with the present lamps. In this case 30 volts could be used, greatly reducing the first cost of storage batteries and also the maintenance cost. The cost of maintaining the white walls and ceilings would have to be considered and would add materially to the expense, cutting down some of the advantages.

Regarding illumination and reflectors as in use at present, 2-ft. candles are ample on the reading page in cars, but owing to various conditions, dust, dirt, low voltage from many causes, and the personality of passengers, it is not safe to figure less than 2½ to 3 candle feet on the reading page with new equipment. Other classes of work in other cars would be proportioned correspondingly.

The Author. The discussion indicates a general agreement that the axle generator system is the most universally applicable of the electric systems described. Its predominating advantage is that it requires no interference whatever with the most favorable utilization of equipment from an operating standpoint. The head-end system (unless all passenger equipment is lighted by this system) does definitely limit the make-up of trains, while the straight storage system sets a definite minimum for layover time, and limits yard movements unless all tracks are equipped with charging facilities. In many cases this freedom of operation is the determining factor in the choice of systems. However, all three are in extensive and successful operation.

Mr. Dixon points out that the paper alludes rather inadequately to the predominating use of Pintsch gas in the car lighting field. This method of lighting was treated very briefly for the reason that all present were familiar with it and the greater part of the paper was devoted to the newer and less familiar electric systems.

Mr. Dixon also calls attention to a new electric system to which items *d* and *f*, Par. 44, do not apply. In the system to which this evidently refers the functions of lamp and generator regulators are performed by a regulating battery which alternates in service with the battery on charge. In another new system inherent voltage regulation is provided in the generator and the battery is

floated on the line in the same manner as stationary batteries are frequently operated. Neither of the systems have been extensively used in this country to date.

Mr. Bliss calls attention to the availability of large axle generators as head-end machines and claims that they have the advantages of head-end steam generators without the disadvantages. It would seem that the principal disadvantage of the steam head-end system is retained, namely, limitation of the make-up of trains.

Mr. Bliss also discusses the relative advantages of taper and zero charge cycles. It is the writer's belief that either system will give perfectly satisfactory service provided the operation of the regulator is positive and accurate; in either case, with the proper setting, the charge is reduced to a negligible current before damage to the battery can result. For the taper charge cycle the setting is somewhat lower and this has the advantage that a slight increase above the normal voltage limit will be less harmful to the battery.

Mr. Young has brought up the question of emergency lighting. Experience shows that with good modern equipments and first class maintenance failures are extremely infrequent and candle holders on such cars as mail cars and diners are a sufficient provision for emergency lighting.

Mr. Young's point that M. C. B. train lighting recommendations should have been mentioned is well taken.

The battery capacity used with the head-end equipment mentioned by Mr. Gilman would give a much lower reserve capacity than is customary for axle systems on Eastern roads.

No. 1381

FACTORS IN THE SELECTION OF LOCOMOTIVES IN RELATION TO THE ECONOMICS OF RAILWAY OPERATION

By O. S. Beyer, Jr., Chicago, Ill.[1]

Non-Member

The problem of locomotive design is comparatively simple when it is clearly known what is desired. The possibility of effecting operating results by the introduction of improved locomotives alone, or by their use in connection with such changes as grade revision, is not as fully appreciated as it ought to be.

2 The wide range of motive power now available will have to be considered in future double track and relocation work, grade reduction, elimination of rise and fall, and curvature and distance, in order to effect the greatest economy possible for the capital expended. It should no longer be necessary when relocating a division to increase its capacity and reduce operating expenses, to go to extremely heavy capital expenditures to reduce grades to the minimum of 0.2 or 0.3 per cent, as it is usually much cheaper to provide locomotives of greater power.

3 The main steps in the careful selection of motive power may be divided into the consideration and study of (*a*) the nature of the business; (*b*) the topography of the road, train speed and train resistances; (*c*) the types and sizes of locomotives available; (*d*) improvements to the permanent plant; (*e*) effect of various types and sizes of locomotives on operating expenses; (*f*) final selection of most economical type and size of locomotive.

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NATURE OF THE BUSINESS

4 The special nature of the business, while intimately related to the general service for which the engines are intended, deserves some particular consideration. Through passenger service, for instance, may vary between long distance high-speed hauls, and slow-speed frequent stops hauls. Freight service, especially, may vary between hauling almost exclusively trains of low class bulk commodities, such as coal and iron ore, which generally move in well built high capacity steel cars at slow speeds, and hauling trains of a decidedly mixed nature, loads and empties, box, furniture, flat, automobile, gondola and stock cars, which are not of high carrying capacity and are seldom fully loaded, resulting in trains of great length when any large amounts of tonnage are hauled. Due to the generally inferior construction of cars which compose a mixed train, the weakness of the draft rigging of the older cars, the limitations of the older types of air-brake equipment, and the tendency to long trains when heavy tonnage is handled on railroads where this class of business is general, the motive power problem assumes an aspect entirely different from that on roads where the business is of the nature first mentioned.

TOPOGRAPHY, TRAIN SPEED AND TRAIN RESISTANCE

5 The consideration of the topography, train speeds, and train resistance of the territory in which the new engines are to operate will at this time only take recognition of hauling capacities and train lengths. The extensive research work recently conducted by many eminent investigators, railroads, and locomotives builders furnishes reliable information and data from which to determine very exactly what the hauling capacities of different types and sizes of locomotives are, and the speeds and resistances of the trains which can be hauled by them.

6 *Passenger Engines.* Grades, speeds, and train resistances must be very thoroughly considered when new passenger engines are to be purchased. Passenger engines are intended for high-speed work. Modern operating conditions frequently present cases of heavy trains composed of all steel cars. The weight of passenger cars, due to the desirability of greater passenger carrying capacity and heavier construction, is rapidly increasing. Hence, the most essential quality to be provided in a

passenger engine is the ability to maintain large sustained tractive efforts at high speeds, as well as high starting efforts at low speeds. This ability of a passenger engine is secured by providing ample boiler capacity and good steam engine efficiency. The ability of a passenger engine to accelerate rapidly is another quality which must be considered, particularly if the engine is to be engaged in service in which stops are frequent. The power required for high rates of acceleration, especially on grades, with heavy passenger trains is enormous.

7 *Freight Engines.* Freight and pusher engines are engaged in a service which requires at the critical time very high tractive efforts at slow speeds. This depends to a large extent upon the weight placed on the drivers and the total weight of such engines must thus be so distributed that a relatively large weight falls on the drivers. With the cylinder and driver dimensions and the boiler pressure so arranged that the maximum tractive effort at the rim of the drivers equals 22 to 25 per cent of the weight on the drivers, a satisfactory combination for an engine for ordinary freight service is secured.

8 Modern conditions demand an increase in fast freight service and relatively large sustained tractive efforts at high speeds over heavy grades are becoming more necessary than ever. Locomotives for fast freight service only may afford to sacrifice some initial tractive effort for the sake of having recourse to a proportionately larger heating surface when great quantities of steam are necessary at high speeds. Pusher engines and road engines, on the other hand, intended exclusively for slow service may be permitted to have a large tractive effort capacity at slow speeds with a sacrifice in high sustained tractive efforts at high speeds.

9 The large majority of freight engines purchased are intended for a service which is a combination of fast freight and drag or slow freight service. Under these circumstances it is usually desirable to get as large a steaming capacity as possible consistent with a high tractive effort at low speeds. Incidentally, engines with good steaming capacities when operated with heavy tonnage trains are proportionately more economical in fuel and water consumption, as well as capable of maintaining a higher average speed over the entire division.

10 Consideration of the topography of the railroad and the hauling capacity of freight locomotives presents the problem

of permissible train lengths. If the territory over which the locomotives are to operate is, to a large extent, a succession of many sags and pitches and contains several momentum grades, the continual running in and out of the slack between the cars is a serious matter and tends to limit the length of the train and the amount of tonnage per train which can be hauled. The nature of the business, whether it is ore, coal or pig iron moving in high capacity modern steel cars, or general merchandise moving in box cars, refrigerators, small capacity gondolas or other similar cars has a further bearing on this feature. The lower the average total car weights of trains the longer the trains when large amounts of tonnage per train are hauled. The introduction of high capacity friction draft gears, steel underframes, and improved air brakes are tending steadily to minimize the difficulty of operating very long trains. The perfection of the variable load brake may help further in this direction. Cases are on record where trains of 90 and even more loaded cars, each car weighing 70 tons, have been hauled with success. Trains of 125 empty cars are not unusual in daily operation. Most of these extremely long trains are running over territories whose grades are low, or which have no broken profile. Furthermore the nature of the lading hauled in these exceedingly long heavy trains, is such that, should any heavy shocks occur, the lading cannot be damaged. Cars filled with automobiles, furniture, or general merchandise or flat cars loaded with agricultural machinery are different and must be handled with greater care in shorter trains. Every district for which motive power is intended presents certain little peculiarities when advisable train lengths are considered. Experience in the operation of trains hauled by the older types of locomotives must assist in determining the final answer to this question. As a rule there are a great many circumstances, such as the capacity of the side tracks and other conditions of the permanent way, which are apt to limit the length of trains before the profile conditions establish limitations.

SIZES AND TYPES OF LOCOMOTIVES

11 *Passenger Locomotives.* The principal types of locomotives available for passenger service are the Atlantic (4-4-2) type and the Pacific (4-6-2) type. The American (4-4-0) type, the Ten-Wheel (4-6-0) type, and the Prairie (2-6-2) type have

been employed to some extent, but except in unusual cases are not of special advantage. The latest development in passenger engines for severe mountain service is the Mountain (4-8-2) type. In some special cases, such as exceedingly heavy mountain service, the Mallet engine has been used.

12 The Atlantic (4-4-2) type locomotive is usually best adapted to a service in which the trains weigh about 300 to 350 tons behind the tender and the grades encountered are relatively light. Owing to its wheel arrangement it permits of a boiler of ample capacity in proportion to the cylinders. It has a short rigid wheel base and a short total wheel base. To get very high initial tractive efforts with locomotives of this type means exceptionally high axle loads, which are undesirable. Hence, when high initial and sustained tractive efforts are required for heavy trains weighing over 350 tons behind the tender and operating over heavy grades, the Pacific (4-6-2) type is usually found to be better. The wheel arrangement of this type permits a still larger boiler, greater total weight on drivers with lower average axle loads, and larger cylinders. In general, the full utilization of these features results in both higher initial and higher sustained tractive efforts, combined with better accelerative qualities.

13 For exceptionally heavy mountain service the Mountain type permits of still larger boiler capacities and greater total weights on drivers, and hence still higher tractive efforts. Several engines of this kind are in service on the Chesapeake and Ohio Railway hauling trains weighing 600 to 650 tons over grades of 70 ft. per mile at 25 to 26 miles per hour.

14 Mallet engines have been introduced to a limited extent in passenger service. The Central Pacific Railroad has 12 Mallets which are hauling passenger trains on grades 116 ft. per mile and 40 miles long. The Atchison, Topeka and Santa Fé Railway has two Mallet compound engines which were built to haul passenger trains. The service in both these cases is very severe and rather exceptional.

15 The Atlantic and Pacific type engines, under modern operating conditions, are, for high-speed and high-capacity passenger service, the most desirable types. Under certain special circumstances, long continuous opposing grades may justify compounding in connection with these engines. The introduction of the high temperature superheater and the sectional brick

arch have helped materially to increase the capacity, fuel economy, and efficiency of passenger engines. The limitations of Atlantic and Pacific type passenger engines are principally controlled by the permissible wheel loads. When 60,000 to 63,000 lb. per pair of drivers is once reached, it is questionable, from many points of view, whether it is wise to go still higher. Hence, when greater tractive efforts are necessary than can be secured from an engine with 180,000 to 190,000 lb. on drivers it becomes a question either of reducing schedules, double heading, or introducing locomotives with an additional pair of drivers.

16 *Freight Locomotives.* Recent developments have made available an exceptional field from which to select locomotives for freight service. It seems limited not so much by the extent to which it is possible to build freight engines as it is by the physical restrictions of the permanent way, the nature of the freight business hauled, length of trains, and topography of the road. These limitations are, of course, mostly very serious, and, as far as track gage is concerned, insurmountable, except perhaps in some special cases.

17 The type of locomotive which has been the standard on many of the American railroads in the past ten years is the Consolidation, or (2-8-0) type. It has been called upon to perform in services ranging from emergency passenger to slow heavy pusher and switching service. Engines of this type are being built for heavy and exacting freight service and their possibilities have not been exhausted. They utilize nearly the total weight of the engine for adhesive purposes. A leading truck of two wheels only is provided permitting of a slightly extended boiler and taking from the drivers only weight enough to secure good guiding qualities. The steaming capacity, firebox size and grate area are necessarily limited, since the entire boiler and firebox must be carried over the drivers. The handicap imposed by the boiler limitations has not, until recently, been very serious.

18 Engines of the Consolidation type, having a maximum tractive power of 60,900 lb. are in service today. The diameter of their drivers is small, 54 in., and their total heating surface compared with the equivalent heating surface of a Mikado engine having the same tractive effort is but 70 per cent. The

piston speeds of these large Consolidation engines, compared with the Mikado engine, are considerably higher.

19 The perfection of the high temperature superheater, the brick arch, and the Gaines combustion chamber opens up further opportunities for the Consolidation engine. The application of the superheater results in increased capacity which corresponds, roughly, to a 25 per cent larger boiler capacity than it was possible to provide in connection with saturated steam engines. The brick arches permit increased amounts of heat to be utilized from the fuel burned on restricted grate areas. It should be possible to build Consolidation engines with good steaming capacities and economical fuel requirements that can develop as high as 54,000 lb. maximum tractive effort.

20 An offshoot from the successful Consolidation freight engine is the 12-wheel or (4-8-0) type. This type has not been widely introduced. It has an undesirable ratio between total weight and adhesive weight. The increase in the length of boiler made possible by the four-wheel truck in place of the two-wheel truck of the Consolidation engine nets but little in the direction of increased boiler capacity. The increase in the heating surface of the boiler is at the wrong end. To improve the steaming capacity of the Consolidation engine it is necessary to introduce modifications at the firebox end.

21 The introduction of such modifications has resulted in the Mikado (2-8-2) type engine. By placing a trailing truck underneath the firebox better boiler construction becomes possible; also a decided increase in effective heating surface, a deeper throat sheet and wider water legs are secured. However, as large a proportion of the total weight of the engine is not utilized for adhesive purposes as with the Consolidation type. By moving the firebox behind the drivers, it also becomes possible to enlarge the boiler diameter, and to increase the relative diameter of drivers, thereby permitting of lower piston speeds. The general construction of the Mikado locomotive is such that it permits of very ample steaming capacity and thus of high sustained tractive efforts. The application of the superheater and brick arch has further increased its capacity in this direction. It is most admirably suited to haul slow maximum tonnage freight trains one day and fast freight trains the next, a condition frequently met in railroad operation.

22 The size of Mikado locomotives for most roads is princi-

pally limited by the allowable weights on drivers. It seems to be generally considered that an individual axle load of 60,000 lb. for the better conditions of roadbed, as they are met with today, is very nearly the largest permissible. If so the Mikado engine, as far as size is concerned, has very nearly reached its limit, and the demand for still larger engines will have to be met either by introducing another pair of drivers, making five pairs in all, or by resorting to the Mallet type. With their weight on drivers limited to about 60,000 lb. per pair, it is possible to build Mikados which have a maximum tractive effort of 60,800 lb. with very favorable sustaining qualities at high speeds. The utilization of the superheater and brick arch in connection with the well proportioned boiler and efficiently designed engine make it possible for this size of locomotive to be operated without requiring excessive amounts of fuel so that one fireman can handle all that is needed.

23 To get still larger capacities than are provided by the Consolidation and Mikado types, the Decapod (2-10-0) and the Santa Fé (2-10-2) types are available. The Decapod (2-10-0) type, like the Consolidation and 12-wheel types, has limitations as regards boiler capacity, in consequence of which it is practically adapted to slow service only. Its high proportion of weight on drivers, giving it a high ratio of adhesion is of advantage for this kind of service. The Santa Fé type permits of better boiler proportions than those of the Decapod type, just as the Mikado is better than the Consolidation. The additional pair of drivers enables a tractive effort about 20 to 25 per cent greater than can be secured from the Mikado engine. Allowing 60,000 lb. per pair, the maximum tractive effort possible should be about 73,000 to 75,000 lb., barring cylinder limitations. Several engines of this type now in service deliver a maximum tractive effort of 71,000 lb. It is reported that they can be handled by one fireman without unduly taxing him.

24 Locomotives with five pairs of coupled wheels have an exceedingly long rigid wheel base. This would introduce many complications should they be placed on territories where track curvature is frequent or severe. Furthermore, the exceptionally heavy pressures on the main pins and the heavy reciprocating parts justify expectation of maintenance difficulties. The long wheel base and the large number of heavy wheel loads in rigid

order may be proportionately harder on the track than is the case with large Mikado engines.

25 Another type of engine which deserves consideration for freight service is the Mountain (4-8-2) type, which is similar to the Mikado in all its characteristics. Where fast freight service is abundant and high speed is frequent the additional advantages in guiding qualities secured by the four-wheel leading truck and the slightly increased boiler capacity are important.

26 The Pacific type engine for exclusive fast freight service, where grades are not severe and where this kind of service is heavy, is a very desirable type. A large number of these engines have been built for this service and are giving an excellent account of themselves.

27 The Mallet type offers quite as wide a field to choose from as the Pacific, Consolidation, Mikado, and Santa Fé types combined. Mallet locomotives have been built on both the compound and the simple principle. The wheel arrangement permits of a great number of practical combinations. The application of the superheater and brick arch, feedwater heater and reheater, together with well proportioned boilers and the compound feature has made possible units of large size and of good drawbar pull characteristics at different speeds. At the same time Mallets are economical in fuel consumption. The arrangement of the drivers in two independent sets, and the division of the total engine weight over these two sets permits readily of meeting track and axle load limitations. Hence these engines offer a large field from which to make selection when the restrictions of the permanent plant are such that they cannot be overcome except by heavy expenditures.

28 Mallet engines can be built to deliver a maximum tractive effort of 140,000 lb. This would mean engines with ten pairs of drivers, each having an average load of about 60,000 lb. As long as 60,000 lb. remains the maximum average practical wheel load, while track curvature remains a consideration, and the gage of the track remains at 4 ft. 8½ in., thereby limiting the height of the center of gravity of engines, it is questionable whether an engine much larger than this can be built. It is not a size which has been reached today, although there are Mallet engines in service which have ten pairs of drivers.

29 A large number of Mallet locomotives are in road and pusher service whose tractive effort working compound range from 73,000 lb. to 105,500 lb. They are meeting with success from the fuel, operating, and maintenance standpoints. The largest number of drivers under the engines referred to is eight pairs, the average weight per pair under the largest one being 58,560 lb. Hence, 105,500 lb. tractive effort is not far from the maximum possible with eight pairs of drivers allowing 60,000 lb. per pair.

30 *Switch Locomotives.* The types of switching locomotives available range from the six-wheel coupled to the ten-wheel coupled. Recently a Mallet compound engine has been placed in hump yard service by the St. Louis, Iron Mountain and Southern Railroad in order that long trains may be handled without breaking them up. Switching locomotives of five pairs of drivers have a rather long rigid wheel base, perhaps too long for the average yard conditions as they exist on many roads today. Locomotives with four pairs of wheels have a more suitable wheel base, and are capable of delivering comparatively high tractive powers. Locomotives of three pairs of drivers are the most universal in service today.

THE PERMANENT PLANT AND ITS RELATION TO MOTIVE POWER SELECTION

31 The permanent plant of a railway as related to the motive power is the track, bridges, passing sidings, terminal yards, engine terminals, including the roundhouses, turntables, coaling stations, watering cranes, ash plant and sanding facilities, and the locomotive repair shops. It has been shown what a wide range of motive power is available from which selections may be made for any class of service. In order that the possibilities of this large field may be fully realized it becomes necessary to study carefully the various changes in the permanent plant to be considered in connection with the introduction of different types and sizes of engines. Such a study will oftentimes show that improvements made to the permanent plant at limited costs will permit of utilizing motive power which will effect a considerable saving in operating expenses, thereby fully justifying the expenditure.

32 *Track.* The improvement of the track with a view of making possible higher train speeds and heavier axle loads is a complicated problem. Track improvements such as laying heavier rails, respacing of ties, increasing the carrying power of the sub-grade by drainage, deepening and improving the nature of the ballast are continually under way. The tendency is to strengthen and improve track as rapidly as earnings will permit, so that full advantage may be taken of all that is offered in the way of possibilities to increase engine capacities and train tonnage. The state of good track today, laid with 90- and 100-lb. rail, is such that it readily allows 60,000-lb. axle loads, which is about as high on the average as it is advisable to go. While the track may be such that, should heavy wheel loads be imposed, it would result in increasing the maintenance costs, the tendency may nevertheless be to improve this lighter track in the near future by introducing ballast and relaying the lighter rail with heavier. In such cases, and they are very prevalent, the present track conditions should not be permitted to limit the motive power sizes too greatly because of the economical advantages of increased train loads.

33 Bridges not only restrict the individual axle load, but also the total weight. Moderate expenditures in the direction of bridge improvement, plus the regular program of bridge renewals and improvements may permit of taking advantage of a type of locomotive which would effect economies in operating expenses that would more than pay for the unusual improvement expenses incurred.

34 *Yards and Passing Sidings.* Unless the yards and passing sidings are adequate to take care of longer trains the improvements expected by the introduction of larger locomotives will not materialize and the operating expenses will show but little decrease. The long trains are apt to be tied up and blockades will occur which are expensive as well as demoralizing.

35 *Locomotive Terminals.* Except for side and overhead clearances, the coal chutes, watering cranes, sanding and ash handling facilities, will have but little effect on the choice of new locomotives. Ash pit construction may need a little modification to permit of heavier wheel loads. The roundhouse may be inadequate to permit of housing the larger engines suitably. The turn-

table may be too short or too light to turn the heavier engines. Limited or inadequate locomotive terminals, in part or in whole, should not be permitted to stand in the way of introducing the most economical locomotive available. It is becoming more generally recognized that as motive power increases in size it becomes relatively more important and economical to provide means at locomotive terminals whereby the turning of power is expedited.

36 *Shop Facilities.* It would be manifestly impractical for a railway to undertake to rebuild its entire shops for the sake of handling a very large engine of the Mallet type. As a rule, however, especially on the larger roads, the present shop facilities are such that a railway, from this point of view, should find but few limitations to the size of locomotives that it seems desirable to operate. Then again the adopted program of shop facility renewals and modifications, continually necessary as business keeps growing and old parts of the shops depreciate, tends to take away from the seriousness of this consideration.

37 *Clearances.* Clearances through cuts, of water cranes, buildings, and other fixed structures along the right of way have some effect in determining the size of the engine to be selected, as well as the amount of the expenditures to be made for improvement.

THE RELATION OF OPERATING EXPENSES TO THE SELECTION OF MOTIVE POWER

38 The effect of the selection of locomotives for passenger and switching service on operating expenses does not play as important a role as it does in the selection of engines for freight service. The choice of passenger and switch engines is determined very largely by imposed conditions resulting from circumstances peculiar to the nature of these two kinds of service. Larger and heavier switch engines are usually made necessary by heavier trains handled in the yards. The demand for larger and improved passenger engines results from the necessity of maintaining high speeds with trains which are growing in weight due to the introduction of steel equipment and more cars. It should be observed, however, that the economy of the various types of engines available, from the fuel, lubricating, and maintenance points of view has an important bearing on the ultimate selection. That type and size of engine should be chosen which

will operate with the least expense for fuel, water, and repairs, and which will keep as low as possible, consistent with the size of the engine in regard to the service requirements for the present and the future, the expeditures for improvement to the permanent plant.

39 In the selection of power for freight service the effect of the various types and sizes on the operating costs should go a long way towards determining the most economical engine to choose. The largest part of the gross revenue of railways results from freight transportation, and the greatest proportion of their operating expenses are consumed in conducting this transportation. The nature of freight service permits a much greater flexibility as far as choice of the motive power is concerned. A study should be made to determine which type and size will effect the greatest net saving in operating expenses after deducting all overhead and additional maintenance charges resulting from the improvements necessitated by the introduction of the engine. Only by such a study as this in conjunction with considering the service conditions and the tendency of future development can the ultimate selection be made with any degree of correctness.

40 The items of operating costs and overhead charges which should be considered in such a study are the following:

a Transportation expenses
- Fuel
- Water
- Lubricants for locomotives
- Other supplies for locomotives
- Enginehouse expenses
- Train supplies and expenses
- Enginemen's wages
- Trainmen's wages

b Maintenance of equipment
- Locomotive repairs
- Freight train car repairs

c Maintenance of way and structures
- Ballast
- Ties
- Rail
- Other track material
- Labor, roadway and track

Bridges, trestles and culverts
Buildings, fixtures and grounds
d Overhead charges
Interest on locomotives and improvements to permanent plant
Depreciation of locomotives and improvements to permanent plant
Taxes and insurance on locomotives and improvements to permanent plant.

41 To attempt to discuss within the limits of this paper the effect of locomotive types and sizes on each one of the items of railway expenses concerned would be impossible. Wellington, Webb, and Henderson have formulated many of the principles involved and reference to the works of these men will reveal considerable valuable data and information on this subject. The statistics kept by the individual railways will furnish additional information necessary for the analysis. The application of logical reasoning and the use of recent data established by both laboratory and service tests will further aid in making correct comparisons. The suggestions regarding the various items of expenses and their relation to motive power selection will therefore be general and seek rather to point out the inherent tendencies of these expenses and their importance.

42 *Fuel.* Fuel is the largest single item of locomotive operating expenses and therefore the most important. The fuel consumption may be divided into the following classes: (*a*) fuel used while actually working on the road; (*b*) fuel used while drifting and waiting; (*c*) fuel used at terminals for firing up. As locomotives grow larger their fuel consumption per unit increases, but not nearly in proportion to the increase in their size. It does not take very much more coal to fire a large locomotive than a small one. The fuel losses of a large locomotive due to radiation while waiting or drifting are but slightly larger than those of a smaller locomotive. The increase of fuel consumption of large saturated simple steam engines when working at their full capacity is more nearly in proportion to the increase in their size. The introduction of the superheater, feedwater heater and reheater, the increase in heating surface of the boiler, the brick arch, the utilization of compounding in large engines of the Mallet type, application of improved valve gear and compound air pumps, and more careful attention to the

design of steam passages and steam engine efficiency have accomplished remarkable results in keeping the fuel consumption of large locomotives down so that their consumption per train-mile is increased but slightly over that of the recent types of smaller saturated steam locomotives.

43 Numerous tests and service records have revealed that large superheater Mikado locomotives which have been placed in service recently haul trains of 45 and 50 per cent greater tonnage with the same amount of coal that was formerly consumed by the Consolidation locomotives they replaced. Even the coal consumption of Mallet engines with grate areas up to 100 sq. ft. has not grown in any way proportionate to the increase in their hauling capacity. Modern engines when running at shortened cut-offs over those portions of the road other than the ruling grades exhibited a still greater economy than when working on the heaviest grades. Some service tests of recently built Mikado engines on the Delaware, Lackawanna and Western Railroad clearly demonstrated these facts. Their economy in fuel consumption as compared with that of the old Consolidation type, both operating over heavy grades at full load, was 20 per cent. The economy effected over easy grades while running at shortened cut-offs was 39.3 per cent, almost twice as much. The average was 29.1 per cent.

44 The conclusions to be reached in regard to the fuel consumption of larger locomotives equipped with those fuel saving devices which have proved their merit is that it increases but slightly as their hauling capacities increase. It depends of course largely upon the size of locomotives in service as to what the actual increase will be on the train-mile basis over the consumption of the old engines, and this must be taken into consideration.

45 *Water.* The same things that are true of fuel consumption, are true of water consumption, only perhaps more so, especially with superheater engines. Water economy increases but slightly with the increase in locomotive sizes.

46 *Lubrication of Locomotives.* It is safe to conclude that the cost of locomotive lubrication increases more or less directly in proportion to the increase in the size of the locomotive as reflected by the wheel and cylinder arrangements.

47 *Other Supplies for Locomotives.* This item is affected slightly, if any, by an increase in locomotive capacities.

48 *Engine House Expenses.* As the size of engines increases the cost of enginehouse expenses per engine handled increases. However, as the size of engines increases the number handled at the terminals decreases. This decrease should approximately counterbalance the increase in cost per engine handled.

49 *Train Supplies and Expenses.* The item of train supplies and expenses per train-mile is affected but slightly by a decrease in train mileage.

50 *Wages of Enginemen.* History clearly shows that the wages of engineers and firemen per trip increase as the engines grow larger. Prompter movements made possible by more powerful locomotives reduce the cost of overtime.

51 *Wages of Trainmen.* The wages of trainmen do not increase per train-mile. If the business is handled a little more promptly through the decrease in the number of trains handled there will be a corresponding decrease in the wages paid for overtime of trainmen.

52 *Locomotive Repairs.* The cost of locomotive repairs does not increase in direct proportion to the increase in hauling capacity. Under modern conditions it will perhaps be safe to assume that about 60 per cent of the total cost of locomotive repairs varies directly as the weights of locomotives increase.

53 *Freight Train Car Repairs.* The item of freight train car repairs per train-mile increases slightly more than in direct proportion to the increase in the number of cars hauled in the longer trains by the larger locomotives. Improvements in car construction, however, are rapidly counteracting this tendency.

54 *Maintenance of Track, Bridges and Buildings.* General observations seem to indicate that about 40 to 50 per cent of the cost of maintaining track per train-mile varies directly with the average increase of weight on drivers. The many improvements in track construction continually under way, the introduction of treated ties, heavier rails of improved steel and better drainage facilities all help to reduce to a minimum the effect of heavier power on the roadbed. The improvements to bridges, buildings and structures, such as roundhouses, turntables and shops, the lengthening of passing sidings and the extension of yard facilities have a further effect on the cost of maintenance of way and structures as related to the increase of motive power sizes.

55 *Overhead Charges.* The overhead charges include inter-

est, depreciation, taxes, and insurance and should be carefully estimated.

FINAL DETERMINATION OF MOST ECONOMICAL LOCOMOTIVE TO ADOPT

56 Taking each one of the items into consideration, estimating the reduction in train mileage effected by each type, the gross savings in operating expenses effected based on the amounts of business on hand or in sight, and deducting therefrom all overhead charges arising from the additional improvements necessary to make the operation of the different types of locomotive under comparison practical, will reveal which particular locomotive is the most economical in size and type.

57 As far as a standard engine of any kind for an entire road is concerned the general conditions obtaining will have some bearing. A road may for instance be composed partly of divisions whose grades are moderate and partly of divisions whose grades are severe. If the variations are not great a compromise standard might be adopted. If on the other hand there is a large difference, it may be wiser to seek to establish two or three standards and confine them to their particular territory with a view to getting the maximum efficiency from every portion of the property. Then again there are many shorter territories such as pusher grades and divisions through mountainous country, the motive power selection for which is a distinctly local problem. In every case, whether it is the broad problem of establishing standards for the entire system, or selecting an engine for a local territory, the problem might well be reduced to an economic study, comparing several available types and sizes, the extent of the improvements necessary to make their operation practical, and the net savings which it is estimated will be effected by their introduction.

SUMMARY

58 In designing new locomotives all of the conditions must first be analyzed and then the design made to suit them. The actual design of the engine finally chosen may be approached with confidence because of accumulated knowledge and experience. Due to the great possibilities of effecting operating results favorably by building locomotives which are exactly suited to their work, a study of the conditions becomes vitally important. To show what these conditions are has been the object of this

paper. The fact that the most powerful locomotives of most approved design are also the most economical should be more generally appreciated. It is to be hoped that the future will see more advantage taken of the modern locomotive in accordance with its possibilities in relation to grade revision and its ability to reduce operating expenses to a minimum. The ultimate benefits which will result will certainly be justified to the fullest extent.

DISCUSSION

H. H. Vaughan said the author was undoubtedly right in his statement that "in designing new locomotives all of the conditions must first be analyzed and then the designs made to suit them," but desired to call attention to another phase of this subject. Steam locomotives had passed through a very rapid development in the past twenty years. The biggest passenger engine at the time of the World's Fair at Chicago was an eight-wheel engine No. 999 which weighed 120,000 lb. About that time the Brooks Locomotive Works designed some heavy power for the Great Northern Railroad and in the years following the World's Fair a great increase in the size of locomotives set in until today 240,000 lb. is quite a moderate weight for a passenger locomotive.

The natural way to increase the size of locomotives is by increasing the cross-section which causes much heavier bearing pressure. This, together with a number of other things, has kept locomotive design in a state of development for a number of years. Larger and larger engines have been built and matters of economy largely neglected. Now it would almost seem as if locomotives had reached a size at which they would remain for a time and that more interest would be taken in matters relating to fuel economy and economy of operation.

On a number of roads there has been a tendency to increase without considering the development of classes of power which could be more or less standardized for the purpose of maintenance and repairs. A great many attempts have been made to standardize miscellaneous classes of engines, which in the speaker's experience is a very expensive matter if done after the engines are built. The time for standardization of engines is before they are purchased and not after they are running.

It would be a desirable thing to establish a standard type of power and then when a change is made move to a decidedly larger

type. When the speaker went to the Canadian Pacific he found that they had in their earlier days been doing exceedingly well at maintaining standard locomotives; but that they were breaking away from this with the introduction of heavier power—getting forty of this class, sixty of that, twenty of that and so on. Fortunately they were able to adopt a different policy and for the last six or seven years have been building engines based upon one standard set of designs. These engines have been constructed of the Consolidation, ten-wheel, and Pacific types, all of which take practically the same small repair parts such as pistons, valves, driving boxes, crossheads, brakes and a majority of the parts that an engine is liable to need for replacement as it passes through its lease of life. There is now a total of 865 engines out of about 1800 or 1900 on the system, all of which are standard engines. For instance, engines of this class have been modified so as to take the Walschaert valve gear instead of the Stephenson and change after change has been introduced as found desirable, but always in such a way that the new arrangement could take the place of the old one for renewals or repairs. Recently a heavier class of power has been designed and that class is now the basis for another series of engines which it is proposed to work out.

This policy has certain advantages. In the first place these engines can be transferred from one part of the road to another without a large storehouse stock. Thus is eliminated the difficulty of moving engines from one part of the line to another on account of there not being material in stock with which to repair them on the division they are moved to. The business of the Canadian Pacific Railway is perhaps more peculiar in this respect than that of other systems because they have a large periodical movement of power from one part of the system to another, and it was only when they got a certain percentage of standardized engines that material with which to make repairs could be provided at all divisional storehouses and the engines could be moved freely all over the road.

In the second place, continued construction of one class of engines reduces the shop cost of material for that class of engine below that of similar repair parts that are of varied design. The fewer the variations in size, design and shop practice, the less difficulty there is in keeping the shop cost down to a satisfactory basis.

In the third place, the construction of standard engines makes it possible to effect desirable variations in design much better than can be done where the classes differ. Where engines are standardized, several divisions are handling them, and many suggestions are received from the divisional officers as to the required alterations on these engines, the design is materially improved.

To obtain standard designs it does not follow that the railroad has to do all the designing itself. On the Canadian Pacific Railway the locomotives were largely designed by the locomotive builders, but to carry on this work successfully a railway must have a good mechanical engineer's office in order to follow it up. The builder often does not realize the effect of small alterations or improvements on the service end of the locomotive, and will sometimes most carefully make slight alterations in the way of improvement, without however going sufficiently into the question of just how this alteration will afterwards affect the replacement of parts. This policy of standardizing power had the disadvantage that each increase in weight of rails or strength of bridges could not be immediately followed by a corresponding increase in the weight of the engine. This disadvantage is small compared to the better service and reduced cost of maintenance obtained from standard power.

Simon Hoffman. The principal advantage to be derived from the application of an efficient superheater to the locomotive is a decrease in steam and fuel consumption and indirectly an increase in the steaming capacity of the boiler, which is equal to a corresponding increase in the hauling capacity of the locomotive.

In speaking of superheaters as applied to locomotives, only such devices are considered as are capable of developing and maintaining a high degree of superheat averaging from 200 deg. to 250 deg. above the saturation point.

The principle which has now been generally adopted in the design of locomotive superheaters is that of a superheater consisting of coils or units disposed in large boiler flues, with forward ends of the superheater units suitably connected with a steam collector casting arranged in the smoke box. The superheater unit pipes go within 2 ft. from the firebox tube sheets and are exposed at their extreme ends to temperatures of from 1400 deg. to 1600 deg., making the superheater capable of developing a superheat of from 200 to 250 deg. Steam of such high

degrees of superheat can be exposed to the cooling action of the steam chest and cylinder walls without condensation and at the same time has about 30 per cent larger specific volume than saturated steam of the same pressure. A large part of this increased specific volume is again lost before expansion of the steam in the cylinders takes place on account of the cooling action of the steam chest and cylinder walls. While the superheat of the steam leaving the superheater may be 200 to 250 deg., the average superheat of the steam in the cylinder at the moment the cut-off takes place is hardly more than 100 deg.; but the entire elimination of all losses through condensation together with the remaining increased volume of the steam effects under average conditions a saving of 30 per cent and more in the steam consumption per indicated horsepower, which gain corresponds to a saving in fuel consumption of from 20 to 25 per cent, compared with a saturated steam locomotive working under the same conditions.

The real value of a locomotive from the motive power point of view must be ultimately measured by the tractive effort which it is able to exert on the drawbar at a certain speed. In order to bring out the bearing the saving in steam and fuel consumption has on the hauling capacity of the locomotive, the following example is given: Assuming two locomotives of the same general dimensions, one being equipped with a high degree superheater, under average working conditions, the superheater locomotive will show an economy in coal consumption of about 20 per cent, if the same indicated horsepower is developed by both engines. If the superheater engine is forced so as to burn the same amount of coal as the saturated steam engine, and assuming that this increased amount of coal can be burned as efficiently in the superheater boiler as in the saturated steam boiler and that the increased volume of steam can be expanded in the cylinders as efficiently as in the saturated steam engine, then the indicated horsepower developed would be $100 \div 80$, or 25 per cent more than in the saturated engine. At the ordinary speeds of passenger locomotives about 70 per cent of the cylinder power is available at the drawbar. Consequently an increase of 25 per cent in indicated horsepower represents an increase in haulage capacity of $25 \times 100 \div 70 =$ approximately 36 per cent if the increased frictional resistance of the running gear caused by the increased horsepower is neglected.

There are two principal requirements for the maximum effi-

ciency of superheater locomotives. The thermal efficiency of the boiler must not be reduced by the application of the superheater. With the type of superheater now generally adopted, no changes are made in the firebox, but the tube heating surface is altered by transferring part of it into the superheater. The superheating surface is not quite as efficient in heat transmission as the evaporating surface, on account of the poor conductivity of superheated steam. In order to make up for this the steam must be forced to pass through the superheater pipes at high velocity even at the risk of wire drawing, and the superheating surface must be so disposed in the gas current, as to offer each cubic inch of gases passing through the superheater more heating surface than the gases find in their passage through the ordinary boiler tubes which are in contact with water. Further, the total gas area through all boiler tubes must not be materially reduced by the application of the superheater and the boiler tubes and flues must be so proportioned and arranged that the stream of combustion gases emerging from the firebox is subdivided so that the necessary amount of gases is diverted through the superheater, in order to furnish the required degree of superheat. These requirements regarding the subdivision of heating surfaces and gas areas were, to the writer's knowledge, first recognized by Dr. Wilhelm Schmidt of Cassel, Germany, who has also had a prominent part in the introduction of highly superheated steam in stationary and marine practice.

It is not always possible for the designer to meet exactly these requirements, but from observation of a great many cases it can be said that under average conditions the total gas area through all boiler tubes is not reduced more than about 5 per cent and the combined total heating surface based on the fire side of the tubes is increased by the same percentage through the application of the superheater. This makes a somewhat sharper draft necessary for the development of the maximum power.

The second requirement assumes that the increased volume of steam be expanded as efficiently, or in other words, that the same cut-offs be used, as in the saturated steam engine. This would mean a corresponding increase in cylinder dimensions which in many cases is not possible on account of limitations in adhesive weight, strength of running gear and other conditions of operation. In switching service, for example, superheater engines make a very favorable showing, although only a moderate degree

of superheat is being developed, but the improvement in efficiency by the application of the superheater is high because the saturated switch engine is extremely inefficient.

All these considerations indicate that the increased hauling capacity of the superheater locomotive is principally caused by the increased steaming capacity of the boiler. It has sometimes been stated that the increased hauling capacity of the superheater locomotive is due to higher mean effective pressure in the cylinders. As a matter of fact, however, in the case of two engines of the same general dimensions, same cylinders and boiler pressures, the mean effective pressure in the superheater locomotive is not only not higher than in the saturated steam locomotive, but sometimes even lower, on account of the expansion line of superheated steam dropping faster than that of saturated steam. Under identical conditions, the superheater engine develops not more power than the saturated steam engine, but the increased steaming capacity of the boiler makes it possible to work the engine at larger cut-offs.

On saturated steam engines in many cases the size of cylinders is limited on account of fear of condensation. The superheater engine is not subject to such limitations and, therefore, allows in many cases a better use of the available adhesive weight. The large modern Mikado type locomotives with their cylinders of 27 in. diameter could indeed hardly work without the superheater, on account of the cylinder condensation which, but for the superheater, would make their operations a practical impossibility.

Repeated tests have been carried on particularly with a view to applying superheaters to old engines with slide valves, and in all cases which came to the writer's knowledge it has been found that slide valves cannot be worked successfully with highly superheated steam. Various means have been developed in order to improve the lubrication of the slide valves, but so far all these trials have failed because highly superheated steam has a tendency to warp the flat valve, which is the principal cause for the rapid wear of its bearing surfaces.

As regards lubrication, the great number of superheater locomotives now in satisfactory service, and causing no more wear in piston and piston valve rings and bushings than saturated steam locomotives, indicates that this question can be successfully solved. Two points, however, should be borne in mind in this connection, i. e., that superheater locomotives, working with

highly superheated steam, should be lubricated with a cylinder oil having a higher flash point than the ordinary cylinder oil used in saturated steam locomotives but without losing its lubricating qualities, and further, means should be provided to protect the oil against carbonization while the engine is drifting, that is, working without steam. This protection can be obtained by the admission of small quantities of steam to the steam chest and cylinders while the engine is drifting after the throttle is closed.

If such auxiliary steam is supplied by an auxiliary steam pipe leading to the cylinders, and worked automatically, i. e., so that the auxiliary throttle opens as soon as the main throttle valve is closed, it is advisable to have some means for protecting the auxiliary valve from being left open while the engine is standing, and admitting steam through the auxiliary steam pipe to the cylinders where it may accumulate and start the engine.

C. C. Trump.[1] While in Germany a year ago I had the good fortune to meet Prof. Johann Stumpf, of the Technical High School, Charlottenburg, near Berlin, and author of a recent book on The Una-Flow Steam Engine, containing photographs, drawings and comparative tests of several locomotives now in service in Russia, Switzerland, Austria and the Prussian State Railways, having una-flow cylinders.

In these cylinders the inlet for steam and the exhaust are separate. Between the inlet and the exhaust the steam flows only in one direction. Steam is admitted at the hot end of the cylinder and is exhausted through ports in the walls of the cylinder controlled by the piston. The inlet end of the cylinder is hot and remains so, and the exhaust belt is cold and remains cold throughout the cycle. The compression is high enough to raise the temperature of the entrapped clearance steam above that of the boiler, with the result that cylinder condensation, even in the case of saturated steam, is practically eliminated.

For simplicity and adaptability these cylinders should meet the requirements of Mr. Vaughan. Only one set of valves is required and the valve gear has actually been applied directly to existing forms, such as the Walschaert. Locomotives abroad have been built with slide valves, with piston valves and with double beat, poppet or lift valves. About 35 una-flow locomotives have been

[1] Humphrey Gas Pump Co., Syracuse, N. Y.

in operation in Europe, several of them as long as two years. The results of a comparative test run between three locomotives of different types may be summarized as follows:

Type of Locomotive (Prussian Railways)	Weight of Train, Tons	Coal Consumption, Tons per 1000 Ton-Kilometers
Stumpf una-flow freight locomotive with superheat	1009	23.7
Lentz freight locomotive with superheat	770	30.6
Improved freight locomotive with saturated steam	770	31.4

These results obtained by recognized authorities show a saving of coal in favor of the una-flow locomotive of nearly 22½ per cent of the consumption of a modern locomotive with superheat.

Other comparative tests have been made and all tend to show that the saving over the best modern types of locomotives is in favor of the Stumpf una-flow engines by at least 10 per cent.

It would seem, therefore, that the una-flow locomotive might be worthy of investigation as an improvement in engine efficiency, which will allow of better economy of coal and increased hauling and speed capacity with a given boiler capacity.

A. W. Gibbs[1] referred to the desirability of carrying the weight on the fewest possible number of axles; or, to put it in another way, of carrying on each axle the greatest permissible weight. He called attention to the fact that while it was customary to compare the severity of different locomotives by considering the static weight per axle, the real figure to consider was the working weight on the rail, which consisted of two parts, viz., (*a*) the static weight, (*b*) dynamic augment caused by the unbalanced vertical component of the counterbalance required to take up the fore and aft action of the reciprocating weights; that this question of the weight of reciprocating parts had received too little attention, and at the high speeds this weight formed a serious addition to the static or scale weight; that the development of certain peculiar forms of rail failures within the past two or three years had attracted attention to the possible maximum pressure which could be permitted between the wheel and the rail, and with a given total maximum it was evident that any refinements in design which would reduce the augment gave just that much

[1] Chief Mechanical Engineer, Penn. R. R. Co., Philadelphia, Pa.

additional freedom in the allowable static weights. It was Mr. Gibbs' belief that an unnecessary number of axles was undesirable, especially in high-speed equipment, but he believed that there was still a considerable margin within which the reciprocating weights could be redesigned, thus increasing the safety margin.

C. D. Young. I cannot agree entirely with Mr. Beyer in his Par. 66 where he states that if the conditions of one or two divisions are analyzed a satisfactory type of locomotive can be selected. I am of the opinion that a compromise design which gives a general purpose engine is the most economical. Specialized designs should be built only when there is a sufficient demand for them in a selected territory and the minimum requirements of this territory should be an indication of the total number of the specialized design which would be selected.

The Pennsylvania Railroad, for example, have a Consolidation type of locomotive considered to be a general purpose locomotive. Its performance as to the boiler and engine efficiency is as good as can be expected from any special design. At least 2000 of that one class of locomotive are on the road. The average railroad man can appreciate what that means in the carrying of stock, the taking care of engine-house repairs, etc., the ease with which power can be transferred from one division to another and the required amount of work to be had from the locomotive.

To illustrate the point further, conditions come up in the summer where a section of the road has a heavy demand for business and it is necessary to transfer power from a division where the work may be heavy in winter or during some other months. If the locomotive is of special design and it is to be transferred to a division where the equipment is of a uniform class, there will be great difficulty for a few months in keeping the engine out of the engine house, and at a time when it is desired to make the heaviest movement.

Mr. Vaughan has brought out a point bearing upon the selection of power, namely, that changes from one design to another should be made very slowly. A change of engine to a more modern class should not be made until it is certain that the weight can be increased, say 5000 lb. per axle. In so doing the number of classes of locomotives to be maintained can be reduced. In other words, it is quite necessary that the maintenance of structure department realize that matters are rapidly progressing and

that the track should be rebuilt and brought to a higher standard so that economy in large train units and larger train loads can be made. When this time is ready there may be sufficient power development to take advantage of the changes. One of the vital problems of locomotive design at the present time therefore is to see how much power can be obtained for the least weight on the wheels; in other words, to provide that the weight on wheels is not in excess of the power requirements.

The Pennsylvania Railroad has under consideration a design which, I believe, is going to make a material saving in the weight of reciprocating parts by a proper selection of treated and alloy steels.

Mr. Hoffman pointed out that as a general rule 1 sq. ft. of superheating surface did not take up as much heat as 1 sq. ft. of water-heating surface. It is desirable to establish what the heat transfer ratio is as between the superheating and water-heating surfaces. A summary of figures obtained at the Altoona testing plant seems to indicate that it lies between 23 and 35 per cent and that it tends to increase between these limits as the power developed increases. If the factor of less than one is correct, then I would like to ask Mr. Hoffman if it is fair to assume that the efficiency of the locomotive boiler for superheating would be the same as the efficiency of the boiler without superheating. It appears to me that it is necessary to give up a certain amount of economy in the boiler due to the application of the superheater, and that this is made up by the economy derived from the superheat in the engine, so that the entire unit, namely, the locomotive, becomes a very much more efficient transportation unit when using superheated steam than when using saturated. In other words, what little economy is lost in the boiler is easily compensated for by the big gain made in the increased engine efficiency. Figs. 1 and 2 are indicator cards taken from the locomotive testing plant. They show in two ways how superheated steam affects the indicator diagram and why we obtain the economy in the engines stated by Mr. Hoffman and also, why we can obtain increased power from the same cylinders when using superheated steam.

In Fig. 1 are shown E-6 locomotive cylinder cards; the cylinders are of the same size, the solid line being a card taken when using saturated steam and the dotted line when using superheated steam. Both cards were taken when the locomotive had

the same cut-off, or about 30 per cent. The expansion line of the superheated steam falls much below that of the saturated steam and this is as would be expected because of the superheated steam being in a less stable condition on account of its higher temperature and much smaller heat content in equal volumes. During the exhaust stroke, the back pressure line for the superheated steam falls below that of the saturated steam, the pressure being about one-half that shown for saturated steam. There is a lower release pressure and consequently a smaller volume of the superheated steam to be discharged from the cylinder, and the pressure is therefore less during the exhaust stroke. The dia-

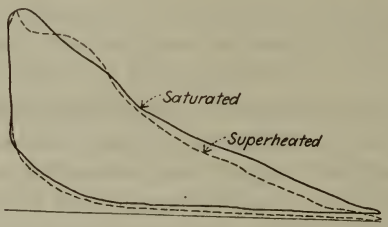


FIG. 1 INDICATOR CARDS FOR SATURATED AND SUPERHEATED STEAM AT THE SAME CUT-OFF

grams show an i.h.p. of about 1500, the superheated steam locomotive using about 10,000 lb. less water per hour and having a water rate of 17.2 lb.

The amounts of heat supplied to the engines per i.h.p-hr. were 22,470 and 29,630 B.t.u. respectively, or the superheater locomotive did the same work at the same speed on a heat expenditure of 24.15 per cent less than that of the saturated steam locomotive. It does not follow, however, that the weights of coal burned in the two locomotives under these conditions are in the same proportion, on account of the slight loss in efficiency of the boiler due to the presence of the superheater. The coal rates for these two tests were 2.19 lb. per i.h.p. for superheated steam and 2.9 lb. per i.h.p. for saturated steam.

In Fig. 2 are shown two sets of indicator diagrams where the water per hour was practically the same for superheated and saturated steam. The cut-off with saturated steam is about 30 per cent and with superheated steam about 50 per cent and the increase in power due to the superheat is over 400 h.p. In these

diagrams the back pressure line for superheat is above that for saturated steam, but in this case the release pressure is much higher with the superheated steam, and the equal weights of water therefore have very different volumes. The increase in horsepower in the first set of tests at 38 mi. per hr. is fully 33.9 per cent for superheated steam. In the second case at 48 mi. per hr. the increase in horsepower is 33.3 per cent.

That the back pressure and compression with superheated

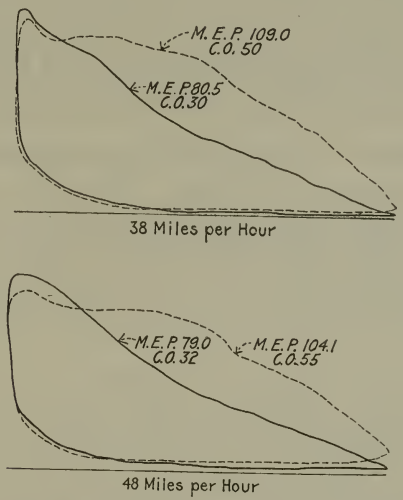


FIG. 2 INDICATOR CARDS FOR SATURATED AND SUPERHEATED STEAM. EQUAL WEIGHTS OF STEAM PER HOUR

steam are so much less than with saturated steam in two engines having cylinders of the same size and otherwise alike, except as to superheater, points clearly to the fact that back pressure may be higher than is necessary to produce the given draft, although since in the superheater locomotive the coal and water consumption is so much reduced, it would be expected that the back pressure for draft producing purposes might be somewhat less for the superheater locomotive.

A large part of the draft is caused by the blast at release, and while the indicator diagram shows approximately the back pressure (i. e., pressure in exhaust pipe) during the return stroke up to the compression point, it does not show the pressure in the exhaust pipe caused by early release from the point of re-

lease to the end of the forward stroke. This must, however, be less than the forward pressure shown by the indicator at release, because of the throttling action of the valve.

Mr. Gaines brings out the subject of tube diameter and firebox volume. This is a very important consideration in the boiler design, but a still more important consideration is the tube length and the ratio of firebox heating surface to tube heating surface. Following the line of thought developed by M. A. Henery of the Paris, Lyons & Mediterranean Railroad, who made some experiments establishing certain tube proportions, I am inclined to believe that the questions of the ratio of tube length to its diameter, and the area of firebox heating surface to total tube heating surface are more vital than the question of firebox volume alone.

G. R. Henderson. Among the various methods of increasing the economy of locomotive operation, feedwater heaters have long been tried in various forms, the simplest probably being the introduction of exhaust steam from the air pump into the tender tank. This has generally been open to the objection, however, that the water is heated to such an extent that injection operation is unreliable.

More recently an extension of the boiler has been converted into a multi-tubular feedwater heater and water from the injector was passed through this section before reaching the boiler proper. As some modern types of locomotives give ample space for a very long boiler, there is no difficulty in providing for such a section to take the heat after it has passed through the boiler flues proper.

Experience with this type of superheater has been somewhat peculiar; in some cases very good results were obtained both in economy and in condition of the heater; but in other cases more or less corrosion has been found in the various portions of the heater, whereas there was no evidence of such corrosion in the boiler proper. These conditions have been more aggravated when good water was used than when water carrying scale and encrusting matter was supplied to the boiler.

In an attempt to overcome this corrosion, in some cases the heater has been practically coupled to the steam space by circulating pipes to both the top and bottom so that it was really an extension of the boiler and would run only partially full of water instead of entirely full as when originally designed as a heater. The strangest part of this problem is that we find when such con-

nections are made the heater does not seem to lose its economical value. There is apparently no increase in fuel required when these circulating pipes are connected with the boiler.

Some builders place a diaphragm sheet a few feet back of the flue sheet and extended as high as the flues. The feedwater is injected into this front section and flows backward over the diaphragm. While we do not know of any comparative tests made to determine exactly the fuel economy of this device, yet it seems as if this arrangement of connecting the heater direct with the boiler so that it practically forms a part of it is likely to be more satisfactory in the long run than when it is used as an individual heater for heating the water in its passage to the boiler without allowing any opportunity for the gases contained in the water to escape as can be done with the circulating pipes.

F. F. Gaines. With increasing wages in all departments, involving also shorter hours and increased overtime, and with the denial of any increase in rates by the Interstate Commerce Commission increased efficiency in every direction is the only method of delaying the closing of the margin between gross income and operating expenses. Greater results can be obtained from efficient motive power than any other item, due to saving in fuel, larger trains for less wages per car-mile, making better time and cutting down overtime, and reduction in the number of trains required to handle a given volume of business which is important in congested territory. Of these various items, however, I will call attention only to the question of fuel economy and boiler capacity, in other words, boiler design.

There have been few radical modifications in boiler design other than the general introduction of superheated steam, except in a few isolated instances. The results obtained by Professor Goss in the Coatesville tests indicate, however, that there should be greater firebox volume and heating surface, and that the ratio of firebox heating surface to tube heating surface should be materially increased. With from 40 to 48 per cent of evaporation from the firebox sheets of the total, it would seem logical to increase firebox heating surface and decrease tube heating surface. The results of the Coatesville tests, while a surprise to many, were anticipated by a few. The Reading Railway has a large number of engines in which the ratio between firebox and tube heating surface is relatively high (one to six) and tubes from about 9 ft.

in length on American type passenger power to only 13 ft. 6 in. on heaviest types of consolidation engines. These results were obtained by the use of the Wooten type boiler with large grate area for anthracite coal, and a vertical brick wall at the rear of the combustion chamber. The brick wall retards the flow of gases and also gives a better opportunity for combustion, as well as changing the path of the gases and forcing them against crown sheet. The combustion chamber furnishes additional firebox heating surface and more volume in which to complete combustion. These engines, while designed for anthracite coal, occasionally ran into soft coal territory and although the grate area was too large for maximum economy the results were excellent.

In consequence of these circumstances a trial was made of arranging a boiler with a suitable grate area for soft coal with a combustion chamber. The back end and one waist course were removed from the original boiler and replaced by a new back end. Carefully conducted tests showed a coal economy of 40 per cent over a sister engine and 15 per cent over the best design of an engine of similar power, but with more liberal heating surface.

In addition to the fuel economy obtained, the shielding of the tubes by the firebox wall has resulted in the engine making over four times the mileage of other engines in the same district before safe ending flues. During this period the engine has seldom, if ever, been reported to have flues blown, while with certain fuel conditions engines in the same territory have required blowing every round trip.

This arrangement of boiler proved so satisfactory that some Pacific type passenger engines were built along the same lines, and carefully conducted tests showed in a flue gas analysis only 0.015 per cent average carbon dioxide, while on a similar engine without the combustion chamber the average was 0.144 per cent.

The most radical improvement that can be made in the locomotive is the use of greater firebox volume and heating surface. This is readily obtained by using a suitable grate area and combustion chamber. It would not be surprising if ultimately a flue length of 16 ft. would become a desirable maximum with 2-in. flues and the combustion chamber substituted for the remaining distance.

W. F. Kiesel, Jr. To determine the possibilities of the Atlantic type of locomotive, the limitations must first be known and

those of paramount importance are governed by the carrying capacity of the track and road bed. Assuming as a basis good road bed, 22-in. spacing of ties and 100-lb. rail, it will be possible to use a weight of 70,000 lb. per pair of drivers. Men from the maintenance of way department will be inclined to say that this figure is too high, but there are possibilities of reducing the weights of reciprocating parts by the judicious use of alloy metals and by careful design, resulting in reduction of counterbalance, which in turn reduces the strain on the rail. One-pound reciprocating weight per 100 lb. of maximum piston pressure is within the bounds of possibilities; for an Atlantic type of locomotive this would permit using counterbalance for reciprocating parts of one-third their weight. At very high speed there would be no more strain on the rail than that induced by heavy locomotives now in service. The load on front and trailer trucks may be taken at 110,000 lb. The total weight of this imaginary locomotive would then be 250,000 lb.

There are few possibilities of improvements in the engine itself. The utilization of heat-treated steel and cast-steel cylinders will reduce the weight, and the weight of the boiler can be increased accordingly. The development of the possibilities of the boiler will bring about improvements of greatest import. The superheater and brick arch have already raised the line between non-stoker and stoker locomotives 50 per cent, leaving all Atlantic type locomotives well within the non-stoker class; therefore, no stoker is necessary. The indications are that great steaming capacity can best be obtained with flues of moderate length, and that the generally accepted theory that long flues increase boiler efficiency is not altogether correct.

As a basis for possible boiler performance the following average figures were obtained from an Atlantic type locomotive on the Pennsylvania Railroad Locomotive Test Plant, at Altoona, Pa. The general characteristics of the locomotive tested are:

Weight	237,000 lb.
Boiler pressure	205 lb.
Cylinders	22 in. by 26 in.
Drivers	80 in.
Fire heating surface	3090 sq. ft.
Grate surface	55 sq. ft.
Flue diameter	2 in. inside, 1¾ in. outside
Flue length	13 ft. 9 in.

The boiler was equipped with a long brick arch, a combustion chamber, and a superheater. Forty tests were made, 8 of 2 hours

each, 9 of 1½ hours each, 17 of 1 hour each, 5 of ½ hour each, and 1 of 15 minutes. The maximum speed was 360 r.p.m. In each test the speed, cut-off, pressure, and power developed were maintained as uniform as possible. In the various tests the average indicated horsepower developed ranged from 756 to 2355. Nineteen tests with average indicated horsepower ranging from 756 to 1700, averaged 2.35 lb. of dry fuel per i.h.p. Thirteen tests with average i.h.p. ranging from 1700 to 2000, averaged 2.71 lb. of dry fuel per i.h.p. Eight tests with average i.h.p. ranging from 2000 to 2355, averaged 3.08 lb. of dry fuel per i.h.p. For this last lot of eight tests the average steam per i.h.p. was 17.03 lb. and the minimum was 16 lb. The maximum evaporation was 38,846 lb. per hr., or 12.6 lb. per sq. ft. of fire heating surface per hour, equivalent to 52,084 lb. per hr. or 16.9 lb. per sq. ft. of fire heating surface per hour from and at 212 deg. fahr.

These tests indicate the possibility of 1 i.h.p. per 100-lb. weight of locomotive, or for $1^1/_3$ sq. ft. of fire heating surface, at 80 mi. per hr.

Returning now to the imaginary locomotive weighing 250,000 lb., the weight on drivers, 140,000 lb., limits the drawbar pull in starting to about 31,000 lb. The boiler can be made sufficiently larger than that of the locomotive tested to furnish steam for 2500 i.h.p. maximum, available for short periods in emergency. For trains of today more than 1800 i.h.p. would seldom be required. At $16^2/_3$ lb. of steam and 2.5 lb. of dry coal per i.h.p., this would require 30,000 lb. of water and 4000 lb. of coal per hour. What can be expected of such a locomotive?

This question can best be answered by determining how much power is required in fair weather to pull ten cars, weighing 600 tons, assuming that the engine and tender together weigh 200 tons. The starting power is just sufficient to start this train of 800 tons on a 1 per cent grade.

The indicated horsepower necessary to maintain a given speed under various grade conditions is given in Table 1:

TABLE 1 POWER TESTS ON ATLANTIC TYPE LOCOMOTIVE

Mi. per Hr.	Per Cent Grade	I. H. P.
40	0.75	2030
40	0.5	1600
50	0.5	2175
50	0.2	1535
60	0.2	2085
60	level tangent	1575
70	level tangent	2160

From these facts it would seem that the possibilities of the Atlantic type locomotive are equal to the requirements of present-day passenger service, for the locomotive weighing 250,000 lb., if given 3300 sq. ft. of effective fire heating surface, would be capable of developing a maximum of 2160 i.h.p. at 40 mi. per hr.; 2280 i.h.p. at 50 mi. per hr.; 2377 i.h.p. at 60 mi. per hr.; 2452 i.h.p. at 70 mi. per hr.; 2510 i.h.p. at 80 mi. per hr.

THE AUTHOR. The development of this subject had from its very nature to be along broad and general lines. It is to be noted with gratification therefore that the discussions have elaborated upon and emphasized the importance of some of the principal considerations of the problem.

The many advantages of motive power standardization pointed out by Mr. Vaughan are all most excellent arguments for a far more careful consideration of the entire problem than it has received in the past. Mr. Hoffman and Mr. Young pointed out the reasons why the superheater had an important bearing on motive power selection, while Mr. Henderson and Mr. Gaines showed the possibilities of the feedwater heater and firebox construction in connection with the subject.

The discussions might further have included some remarks about the locomotive stoker. This device will undoubtedly have a growing influence on motive power selection. While it is a fact that the superheater, the brick arch, improved boiler construction, better engine proportions and more scientific methods of firing, have extended the limits for size of non-stoker engines, the indications are, however, judging from the operating economies effected by the large locomotives made possible by the improvements just mentioned, that still larger locomotives, made possible by the stoker, will effect still greater economies. The locomotive stoker has been developed and perfected along more than one line so as to permit its application with confidence where necessary today.

In its ultimate analysis the problem of motive power selection is entirely an economical one. Were all the available information properly gathered and compiled, it would still be found that a great deal is lacking to solve each problem that comes up. It is with pleasure therefore that those interested in railway operation note that there are now several locomotive test laboratories in active operation in the United States, from all of which new and valuable information upon the selection and design of motive power can confidently be expected.

No. 1382

THE STRENGTH OF GEAR TEETH

By Guido H. Marx, Stanford University, Cal.

Member of the Society

The teeth of gear wheels when transmitting power are individually subjected to an action akin to that applied to a beam fixed at one end, with a load somewhere between the fixed and the free ends. All standard formulae or diagrams for the proportioning of such teeth therefore involve a factor representing the allowable unit fiber stress in a cantilever beam subjected to a bending moment.

2 The experiments described in this paper were undertaken with the primary purpose of throwing some light, if possible, upon the question of this allowable unit fiber stress for cast-iron gear teeth under operating conditions, since definite data upon this point have been lacking, particularly with reference to the effect of pitch line velocity. To this vital lack, attention has been called repeatedly by writers upon the subject of gearing.

3 Thus, in Wilfred Lewis's well-known Investigation of the Strength of Gear Teeth[1] occurs the following statement bearing upon this matter:

> What fiber stress is allowable under different circumstances and conditions cannot be definitely settled at present, nor is it probable that any conclusion will be acceptable to engineers unless based upon carefully made experiments. In the article referred to[2] certain factors are given as applicable to certain speeds and, in the absence of any later or better light upon the subject, Table II has been constructed to embody in convenient form the values recommended.

[1] Proceedings of the Engineers' Club, Philadelphia, 1893, vol. 10; also American Machinist, May 4, 1893.

[2] Power Transmitting Mechanism: On the Strength of the Teeth of Wheels, J. H. Cooper, Journal of the Franklin Institute, 1879.

Presented at the Annual Meeting 1912, of The American Society of Mechanical Engineers.

TABLE II SAFE WORKING STRESS s FOR DIFFERENT SPEEDS

Speed of Teeth, Ft. per Min.	100 or less	200	300	600	900	1200	1800	2400
Cast Iron.....	8000	6000	4800	4000	3000	2400	2000	1700
Steel.....	20000	15000	12000	10000	7500	6000	5000	4300

It cannot be denied that slow speeds admit of higher working stresses than high speeds, but it may be questioned whether teeth running at 100 feet a minute are twice as strong as at 600 feet a minute, or four times as strong as the same teeth at 1800 feet a minute. For teeth which are perfectly formed and spaced, it is difficult to see how there can be a greater difference in strength than the well-known difference occasioned by a live load or a dead load, or two to one in extreme cases. But, for teeth as they actually exist, a greater difference than two to one may easily be imagined from the noise sometimes produced in running, and it should be said, that this table is submitted for criticism rather than for general adoption.

4 It is very evident that Mr. Lewis only offered the values of his Table II tentatively. A careful examination of Mr. Cooper's paper fails to disclose any table from which Table II is immediately transferred, but on page 15 of that paper will be found a series of factors, credited to E. R. Walker, New Castle-under-Lyme, 1868, varying with the rim-speed of wheels, by which the "breaking load of tooth" is to be divided. These factors are given as follows:

m = 3 for very slow speed without shock.
= 4 when rim of wheel runs 3 ft. per sec.
= 5 when rim of wheel runs 5 ft. per sec.
= 6 when rim of wheel runs 10 ft. per sec.
= 8 when rim of wheel runs 15 ft. per sec.
= 10 when rim of wheel runs 20 ft. per sec.
= 12 when rim of wheel runs 30 ft. per sec.
= 14 when rim of wheel runs 40 ft. per sec.

It will be seen that these values correspond to Mr. Lewis's table if 24,000 and 60,000 lb. per sq. in. are used for the ultimate fiber stress in flexure of cast iron and steel respectively. This is an apparent oversight on Mr. Lewis's part. While 24,000 lb. per sq. in. is a legitimate assumption for the unit strength of cast iron in tension, a value of at least 36,000 may be taken ordinarily for the modulus of rupture for transverse stress or ultimate unit stress in outer fiber, due to bending. That Mr. Cooper meant this modulus of rupture by "breaking stress" is obvious from pages 8 to 11 of his paper. He also quotes J.

Christie as using a modulus of rupture of 36,000, with a factor of safety of never less than 4, and then under the most favorable conditions, as well-fitted gears, rigidly supported, running at moderate speed, and stress evenly distributed. For strains suddenly applied the factor of safety should be 6, and when accompanied by severe shocks and sudden reversions of strains it should be 8. With these assumptions, for cast-iron teeth, Mr. Christie would allow for safe working stresses from $\frac{36000}{4} =$ 9000 lb. per sq. in., down to $\frac{36000}{8} = 4500$ lb. per sq. in., as contrasted to a range in Table II of from 8000 lb. down to 1700 lb.

5 In view of the fact that Mr. Christie gives several examples from practice to sustain his position, and of Mr. Lewis's own doubt as expressed in the foregoing quotation from his paper, of so wide a range being justified as that called for by his Table II, and of the further fact that this table was based upon a modulus of rupture much too low, it seems strange that these tentative values have been so generally accepted; and the need of adequate experimental data becomes apparent.

6 Mr. Cooper on page 9 of his paper says:

> It is certainly necessary to know the strain that breaks a piece of projecting cast-iron of given size and shape, when the stress is laid on quietly, as well as when it is driven on with considerable velocity.
>
> It is also important to know the fractional part of the rupturing weight or stress which may be repeatedly laid on with perfect safety, to insure the continuance of adhesion under the usual conditions of working. Upon these essential features of each case, the criteria, almost every engineer's guide book is silent.

7 A copy of Brown & Sharpe's Treatise on Gearing which the writer has had since 1892 gives the only data he recalls having seen of cut gears tested to rupture under running conditions. On page 118 is the following:

> We give a few examples of average breaking strain of our Combination Gears, as determined by dynamometer pressure taken at the pitch line.

Diametral Pitch	Face	No. of Teeth	R. P. M.	Pressure at Pitch Line
10	1 1/16	110	27	1060
8	1 1/4	72	40	1460
6	1 9/16	72	27	2220
5	1 7/8	90	18	2470

If we take a safe pressure at $^1/_3$ of the foregoing breaking strain we shall have

for 10 Pitch $353^1/_3$ lb. at the pitch line
8 Pitch $486^2/_3$ lb. at the pitch line
6 Pitch 740 lb. at the pitch line
5 Pitch $823^1/_3$ lb. at the pitch line

8 In order to add to the available data concerning the strength of modern cut cast-iron gear teeth under operating conditions, the writer had the apparatus shown in Fig. 1, constructed. It consists of a base-plate carrying three adjustable bearings. Only one of these is clearly seen in the immediate foreground of the illustration, but at its right can be seen a corner of a second, while the third is hidden by the brake wheel. The action of the apparatus, reduced to the utmost simplicity of construction, can be seen by reference to Fig. 2.

9 The motor, a 25-h.p. 220-volt, direct-current machine, is connected by a Morse chain to shaft No. 1. Shafts No. 1 and No. 2 are connected by a pair of cut-steel change gears of 8 diametral pitch which can be varied to give a wide range of velocity ratios. Shafts No. 2 and No. 3 are connected by the cast-iron, cut gears to be tested. On the outer end of shaft No. 3 there is a flanged, water-cooled brake wheel carrying a prony brake. The arm of the brake rests, by means of a steel knife edge and plate, on a platform scale. Each bearing is provided with a sight feed lubricator. The measurement of the efficiency of transmission not being the object of the experiment, the friction of the bearings is neglected. The only effect of this is to make the computed breaking load of the teeth a very little less than its real value in each case By means of the slotted base-plate and the tongues on the bottom of the bearings, the latter can be slid into place, the gears accurately meshed without binding or backlash, and then securely held by means of two square-head bolts to each bearing. The Morse chain was lubricated with

a graphite and grease mixture, and the gears, both steel and cast-iron, with ordinary thick grease lubricant which was also freely used on the brake. Before starting each run a wooden guard was placed over the gears to be tested. That this was not a useless precaution was indicated by the fragmentary condi-

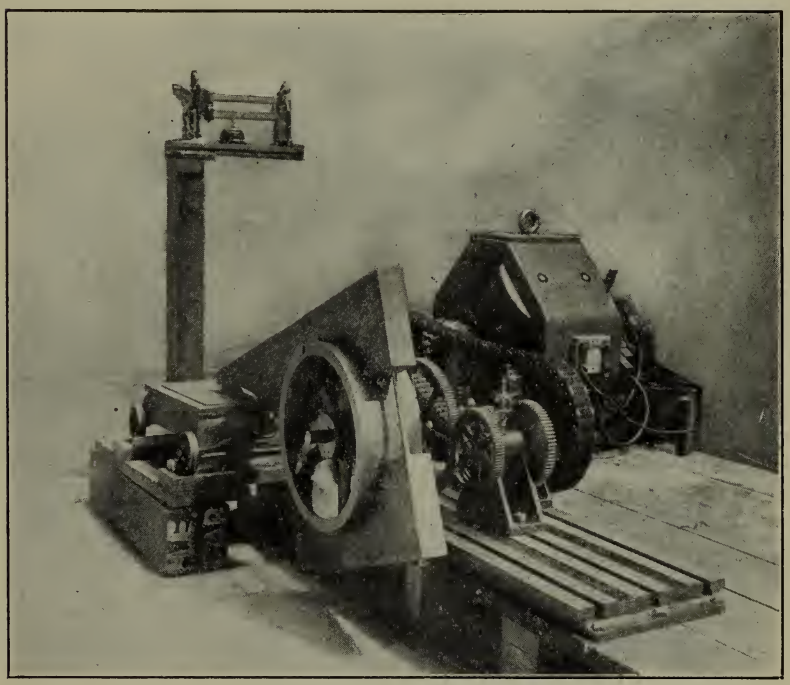


FIG. 1 GENERAL VIEW OF TESTING APPARATUS

tion of the gears, particularly those having arms, at the conclusion of many of the runs.

10 The tests were made in the mechanical laboratory of the Leland Stanford Junior University. For each run the motor was started with zero load applied by the prony brake. The scale weights were then set at the lowest load and the brake tightened until the scale beam floated. Simultaneously the rate of rotation was observed with a tachometer which was calibrated upon completion of the tests and the necessary slight corrections made in the computed results. Calibration of the scales showed them to be accurate throughout the range used. Incre-

ments of load on the scale began at 5 lb. each while the load was low, and were diminished to 2 lb. and 1 lb. as the probable breaking load was approached. The unexpectedly high breaking strength shown by the teeth,[1] particularly at the higher velocities, made it impossible to break the gears at pitch speeds much exceeding 500 ft. per min. with this apparatus.

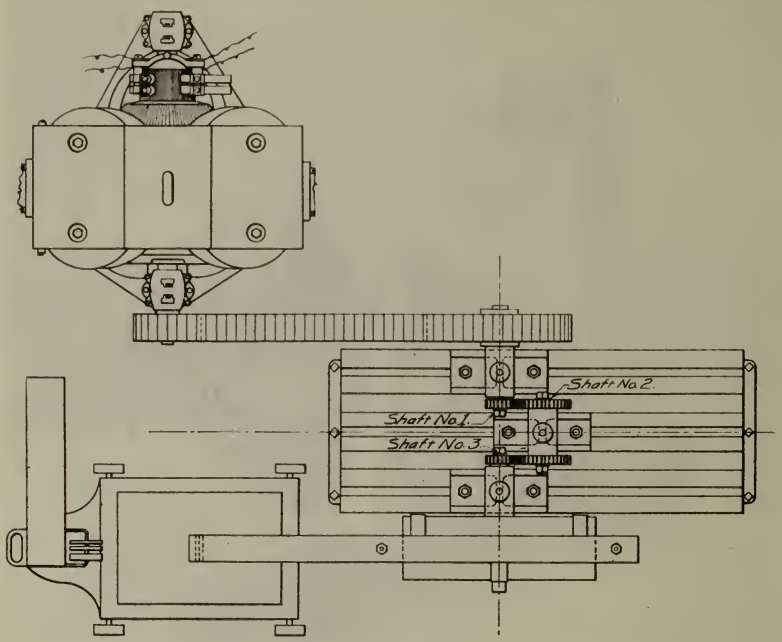


FIG. 2 PLAN OF APPARATUS

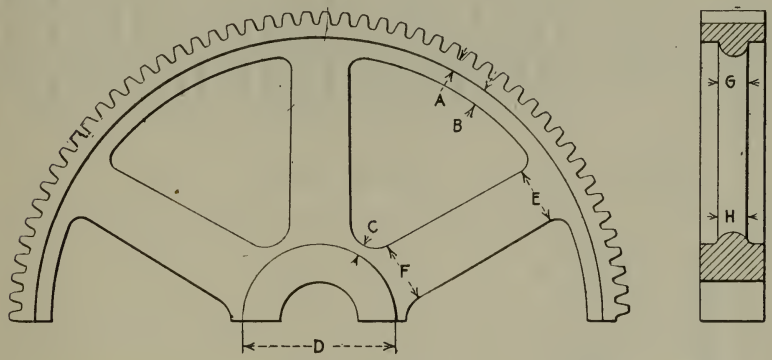
At the higher speeds the teeth were capable, without rupture, of transmitting all of the power the motor could develop.

11 The gears tested were all 10 diametral pitch, cast-iron, 14½-deg. involute, purchased of the Brown & Sharpe Manufacturing Company without intimation of the special purpose for which they were intended, and were of the ordinary stock proportions. The width of face in all cases was 1 1/6 in. The 20- and 30-tooth gears were solid, the 40-tooth webbed, and the others each had six arms. The twenties and thirties had a bore of 1 1/6 in.

[1] Preliminary calculations had been based upon the Brown & Sharpe figures quoted in Par. 7.

and a ¼-in. keyway. The rest had a bore of 1 5/16 in. and a 5/16-in. keyway. The general proportions of the arm gears are shown in Fig. 3.

12 As shown by the logs, the main tests were conducted in two series; the first being made in 1911 with single observations under each set of conditions; and the second, made in 1912, being planned to cover the ground more completely, with the further intention of making three runs in each case under identi-



	A	B	C	D	E	F	G	H
60 T	1/4	1/4	1/4	2 7/16	7/8	13/16	3/8	3/8
80 T	9/32	1/4	1/4	2 7/16	7/8	15/16	3/8	3/8
100 T	5/16	3/16	3/16	2 1/2	7/8	1	7/16	15/32
150 T	3/8	9/32	5/16	2 1/2	31/32	1 3/16	1/2	17/32

Note-All dimensions are in inches. Bore 1 5/16 in. Face 1 1/16 in. in all cases

FIG. 3 DIMENSIONS OF ARMS OF GEARS TESTED

cally the same conditions in order to eliminate errors of observation and of variable material.

13 Fig. 4 shows graphically the results of all the foregoing tests involving 20-tooth pinions. Abscissae are pitch-line speeds in feet per minute, and ordinates are forces at the pitch point equivalent to the breaking loads in pounds. This equivalent breaking load is, of course, equal to the net brake load at rupture multiplied by brake arm divided by the pitch line radius of gear on brake shaft. Curve *A* is drawn representing the results of all tests with 20- and 30-tooth gears in mesh. As there were several observations taken for each pitch speed, the numerical average

TABLE 1 TESTS OF 10 DIAMETRAL PITCH, CAST IRON, 14½-DEG. INVOLUTE GEARS
LOG OF FIRST SERIES OF RUNS

Test No.	Date, 1911	Gears Tested 10 Pitch, C. I.		Sprocket r.p.m. Observed Tacho-meter	Brake r.p.m. Observed Tacho-meter	Change Gears 8 Pitch, Steel		Final Net Brake Load, Lb.	Notes made at Conclusion of Each Test
		Driver	Driven			Sprocket Shaft	Inter-mediate Shaft		
01	June 5	20	150	...	260	120	20	110	Neither gear broken. Fuse blew and motor stopped. Intermediate shaft heated and seized; driven out, draw-filed; bearing scraped. The 5/16 in. machinery steel key of 120–T, 8 pitch, steel change gear sheared off. Shaft replaced by new one in subsequent tests. Both 5/16-in. keys replaced by 1/2 in. This was trial run to test out apparatus.
02	June 9	20	100	...	375	120	20	125	Neither gear broken. Limit of motor power. Fuse blew.
03	June 9	20	100	355	...	20	120	240	Broke 3 teeth at root in 20–T pinion. Broke 100–T gear in 4 pieces. No teeth broken out of large gear, but rim broken.
04	June 9	20	30	353	...	20	120	71	Broke as load was being increased from 70 to 72 lb. Broke 20–T pinion in 3 pieces, each running to the bore; no teeth broken at root. Apparently gave way at key-way. Broke 1 tooth at root in 30–T gear.
05	June 10	20	30	316	...	70	70	50	Broke 13 teeth at root in 20–T pinion. Also broke at key-way and into 3 pieces. Broke 8 teeth out of 30–T gear. Battered and loosened 1/4-in. key on intermediate shaft. Both 1/4-in keys replaced by hardened tool steel, same size. Probable that break occurred by "canting" of loosened key.
06	June 10	20	30	...	360	70	40	60	Broke 20–T pinion in 4 pieces, one at key-way. One tooth cracked at root. No break in 30–T gear. Broke as load was beginning to be raised from 60 lb. to next scale setting, 65 lb.

07	June 10	20	30	...	480	70	30	55	The 30–T gear was the same one used in test No. 06. Broke 20–T pinion in 4 pieces, one being a single tooth with wedge running to key-way. Broke 7 teeth off at root. Also broke 7 teeth off at root in the 30–T gear.
08	June 12	20	30	...	218	70	70	60	This test was made as a repetition of No. 05, as that test seemed unreliable because it pointed to splitting by loose key. Broke 20-T pinion in 2 pieces; also broke out 10 teeth at root and cracked 2. Broke 3 teeth out of 30–T gear.
09	June 12	30	40	...	236	70	70	79	Broke 18 teeth of 30–T gear at root and crushed one from end. Broke 14 teeth at root of 40–T gear. Heard crack at 78 lb. load and stopped run. No teeth found broken. Started again and broke between 78 and 80 lb. load.
010	June 12	40	60	...	206	70	70	148	Broke 9 teeth out of 40–T gear and 9 teeth out of 60–T gear.
011	June 12	40	40	...	230	70	100	104	17 teeth stripped and 7 partially broken on 40–T driver. 10 teeth stripped and 2 partially broken on 40–T driven.
012	June 12	40	60	335	...	70	100	148	9 teeth stripped and 1 broken on 40–T gear. 6 teeth stripped out of 60–T gear.
013	June 12	40	80	326	...	70	100	222	8 teeth stripped out of 40–T gear. Note depth of break at bottom. Broke rim of 80–T gear in 4 pieces.
014	June 12	60	80	328	...	70	100	223	Broke between 222 and 224 lb. load. 21 teeth stripped out of 60–T gear. 80–T gear broken in 6 pieces and 8 teeth stripped.
015	June 13	80	100	315	...	40	100	280	No teeth broken out of either gear. 100–T gear broke at 3 places in rim, also broke out hub and arms at key-way. 5/16-in. key loose in key-way. It looks as if the yielding of the key-way in shaft allowed the key to cant slightly and split the hub. (See also tests Nos. 05 and 016). Key replaced with hardened tool-steel key. Strength of teeth not shown in this test.
016	June 13	30	150	328	...	40	100	380	30–T gear had 7 teeth stripped. 150–T gear very obviously wedged apart from hub outward by the key which was found canted so that one corner of the bottom was 5/64 in. from the bottom of key-way. (See tests Nos. 05 and 015). Large gear cracked at hub and rim, not broken apart.

TABLE 2 TESTS OF 10 DIAMETRAL PITCH, CAST-IRON, 14½-DEG. INVOLUTE GEARS
COMPUTED RESULTS OF FIRST SERIES OF RUNS

SPEED CORRECTIONS MADE FOR ERROR OF TACHOMETER; BRAKE ARM = 32 IN.; ZERO BRAKE LOAD = 3½ LB., BALANCED ON UPPER SCALE-BEAM

Test No.	Teeth in Gears				Correct r.p.m. Brake Gear	Pitch Circumference in Ft. Brake Gear	Net Brake Load in Lb.	Pitch Speed Brake Gear, Ft. per Min.	Equivalent Load at Teeth, in Lb.	Remarks
	Change Gears		Tested Gears							
	Sprocket 8 Pitch, Steel Driver	Intermediate 8 Pitch, Steel Driven	Intermediate 10 Pitch, C. I. Driver	Brake 10 Pitch, C. I. Driven						
01	120	20	20	150	266.0	3.927	110	1044.6	469	Gears not broken. Motor stalled. Run does not count.
02	120	20	20	100	384.0	2.618	125	1005.3	800	Gears not broken. Motor stalled. Run does not count. Demonstrates that teeth are stronger than this.
03	20	120	20	100	12.13	2.618	240	31.76	1536	Reliable run.
04	20	120	20	30	40.22	0.7854	71	31.61	1515	Reliable run.
05	70	70	20	30	215.3	0.7854	50	169.23	1067	Indicates that gear broke as suspected (see Log, Table 1) by canting of key. Repeated in test No. 08
06	70	40	20	30	369.0	0.7854	60	290.03	1280	Reliable run.
07	70	30	20	30	493.0	0.7854	55	387.5	1173	Reliable run
08	70	70	20	30	223.0	0.7854	60	175.38	1280	Reliable run; takes place of test No. 05, of which it was a duplicate but with new key
09	70	70	30	40	241.0	1.047	79	252.33	1264	Reliable run
010	70	70	40	60	210.5	1.571	148	330.7	1579	Reliable run
011	70	100	40	40	235.0	1.047	104	246.05	1664	Reliable run
012	70	100	40	60	160.7	1.571	148	252.46	1579	Reliable run
013	70	100	40	80	116.9	2.094	222	244.79	1776	Reliable run
014	70	100	60	80	176.4	2.094	223	369.38	1784	Reliable run
015	40	100	80	100	103.0	2.618	280	269.65	1792	No teeth broken. Shows these teeth are stronger than even this
016	40	100	30	150	26.9	3.927	380	106.64	1621	Indicates further that gear broke (see Log, Table 1) by canting of key

TABLE 3 TESTS OF 10 DIAMETRAL PITCH, CAST-IRON, 14½-DEG. INVOLUTE GEARS
LOG OF SECOND SERIES OF RUNS

Test No.	Date, 1912	Gears Tested 10 Pitch, Cast Iron		Motor r.p.m. Observed Tachometer	Brake r.p.m. Observed Tachometer	Change Gears 8 Pitch, Steel		Final Net Brake Load, Lb.	Notes made at Conclusion of Each Test
		Driver	Driven			Sprocket Shaft	Intermediate Shaft		
1	May 4	20	30	420	...	20	120	68	20-T pinion broke at key-way; 3 pieces, and 1 tooth broken out at root. Broke 1 tooth out of 30-T gear
2	May 4	20	30	422	...	20	120	78	20-T pinion broke at key-way; 3 pieces, and 1 tooth broken out at root. Broke 1 tooth out of 30-T gear
3	May 4	20	30	422	...	20	120	76	20-T pinion broke at key-way; 3 pieces, and 1 tooth broken out at root. Broke 1 tooth out of 30-T gear
4	May 4	20	30	410	...	70	70	62	20-T pinion broke at key-way; 3 pieces, and 14 teeth broken out at root. Broke 5 teeth out of 30-T gear
5	May 4	20	30	414	...	70	70	58	20-T pinion broke at key-way; 5 pieces, and 4 teeth broken out at root. Broke 5 teeth out of 30-T gear
6	May 4	20	30	414	...	70	70	60	20-T pinion broke at key-way; 3 pieces, and no teeth broken out at root. 30-T gear not broken at all
7	May 9	20	30	450	...	100	40	52	Neither gear broke. Fuse blew. Motor stalled. Made new electrical connections at higher voltage
8	May 11	20	30	...	520	100	40	52	Same gears as test 7. 20-T pinion broke at key-way; 5 pieces, and 1 tooth broken out at root. 30-T not broken
9	May 11	20	30	...	540	100	40	58	20-T pinion broke at key-way; 5 pieces, and 1 tooth broken out at root. 30-T gear not broken
10	May 11	20	30	...	550	100	40	58	20-T pinion broke at key-way; 4 pieces, and 1 tooth broken out at root. 30-T gear not broken
11	May 11	20	30	...	660	120	40	54	20-T pinion broke at key-way; 4 pieces, and 1 tooth broken out at root. 30-T gear not broken
12	May 11	20	30	...	660	120	40	60	20-T pinion broke at key-way; 4 pieces. No teeth broken out of either gear. Note strength of teeth indicated

TABLE 3—*Continued*

Test No.	Date, 1912	Gears Tested 10 Pitch, Cast Iron		Motor r.p.m. Observed Tacho-meter	Brake r.p.m. Observed Tacho-meter	Change Gears 8 Pitch, Steel		Final Net Brake Load, Lb.	Notes made at Conclusion of Each Test
		Driver	Driven			Sprocket Shaft	Inter-mediate Shaft		
13	May 11	20	30	...	660	120	40	52	20–T pinion broke at key-way; 4 pieces, and 3 teeth broken out at root. 30–T gear not broken
14	May 11	20	30	680	...	120	30	48	Did not break gears. Too fast for brake. Abandoned attempt to make runs at this speed
15	May 11	20	30	...	400	70	40	52	Same gears as test 14. Broke before the scales quite balanced at 52 lb. setting. 20–T pinion broke at key-way; 4 pieces, and 1 tooth broken out at root. 30–T gear not broken
16	May 11	20	30	...	396	70	40	54	20–T pinion broke at key-way; 4 pieces, and 1 tooth broken out at root. 30–T gear not broken
17	May 11	20	30	...	400	70	40	56	20–T pinion broke at key-way; 2 pieces, and 15 teeth broken out at root. 30–T gear 1 tooth at root. 56 lb. load barely reached
18	May 11	30	40	700	...	20	120	116	30–T gear 14 teeth out at root. 40–T gear 3 teeth out at root
19	May 11	30	40	720	...	20	120	120	30–T gear 8 teeth out at root. 40–T gear 3 teeth out at root
20	May 11	30	40	710	...	20	120	123	30–T gear 11 teeth out at root. 40–T gear 3 teeth out at root, and 1 broken
21	May 11	30	40	715	...	70	100	112	30–T gear 13 teeth stripped and 7 broken. 40–T gear 6 teeth stripped, 2 broken, 2 cracked at root
22	May 15	30	40	690	...	70	100	108	30–T gear 18 teeth stripped. 40–T gear 8 teeth stripped

23	May 15	30	40	720	...	70	100	106	30–T gear 11 teeth stripped and 1 broken. 40–T gear 9 teeth stripped
24	May 15	30	40	720	...	70	70	102	30–T gear 28 teeth stripped and 1 broken. 40–T gear 15 teeth stripped
25	May 15	30	40	720	...	70	70	102	30–T gear 11 teeth stripped and 2 broken. 40–T gear 9 teeth stripped
26	May 15	30	40	720	...	70	70	103	30–T gear 16 teeth stripped and 1 broken. 40–T gear 15 teeth stripped and 1 broken
27	May 15	30	40	720	...	70	40	92	Neither gear broken. Circuit breaker out. Motor stopped
28	May 15	30	40	710	...	70	40	98	30–T gear had all teeth stripped. 40–T gear 14 teeth stripped and 1 broken
29	May 15	30	40	710	...	70	40	96	Neither gear broken. Circuit breaker out. Motor stopped
30	May 15	30	40	680	...	70	40	100	Neither gear broken. Circuit breaker out. Motor stopped
31	May 15	30	40	680	...	70	40	102	Neither gear broken. Circuit breaker out. Motor stopped. No use trying to break at this speed. Limit of motor. Same gears used in tests 29, 30 and 31; practically one run
32	May 15	30	40	720	...	100	70	108	Same gears used as in tests 29, 30 and 31. 30–T gear had 24 teeth stripped and 1 broken. 40–T gear had 19 teeth stripped
33	May 15	30	40	720	...	100	70	104	30–T gear had all teeth stripped. 40–T gear had 21 teeth stripped and 3 broken
34	May 15	30	40	720	...	100	70	106	30–T gear had all teeth stripped. 40–T gear had 19 teeth stripped. 106 lb. load barely reached
35	May 15	30	60	720	...	70	70	154	30–T gear had 16 teeth stripped. 60–T gear had 8 teeth stripped
36	May 15	30	60	720	...	70	70	156	30–T gear had 16 teeth stripped and 2 broken. 60–T gear had 5 teeth stripped, 1 broken
37	May 15	30	60	720	...	70	70	150	30–T gear had 20 teeth stripped. 60–T gear had 6 teeth stripped
38	May 16	30	80	710	...	70	70	202	Intermediate bearing slipped, allowing load to come on ends of teeth. 30–T gear stripped 6 teeth. 80–T gear stripped 2 teeth
39	May 16	30	80	725	...	70	70	204	30–T gear stripped 5 teeth. 80–T gear stripped 1 tooth; broke into 3 pieces, split at key-way

TABLE 3—*Continued*

Test No.	Date, 1912	Gears Tested 10 Pitch, Cast Iron		Motor r.p.m. Observed Tachometer	Brake r.p.m. Observed Tachometer	Change Gears 8 Pitch, Steel		Final Net Brake Load, Lb.	Notes made at Conclusion of Each Test
		Driver	Driven			Sprocket Shaft	Intermediate Shaft		
40	May 16	30	80	720	...	70	70	206	30–T gear not broken at all. No teeth broken out of 80–T gear; broken into 2 pieces, split at key-way. Key-way found enlarged and key had to be reset by calking more than usual amount
41	May 16	30	100	720	...	70	70	250	Same 30–T gear as used in test 40. Again not broken at all. No teeth broken out of 100–T gear, which broke into 4 pieces; split at key-way. Not necessary to try any more of these as they split at key-way. Try later the "German" key
42	May 16	30	30	720	...	70	70	72	Same 30–T gear as driver as used in tests 40 and 41; stripped 20 teeth and broke 1. Driven 30–T gear stripped 26 teeth
43	May 16	30	30	720	...	70	70	70	Driver 30–T gear stripped 24 teeth and broke 1. Driven 30–T gear stripped 24 teeth
44	May 16	30	30	720	...	70	70	68	Driver 30–T gear stripped 21 teeth and broke 1. Driven 30–T gear stripped 23 teeth
45	May 21	30	80	720	...	70	70	198	"German" key used. Same 30–T gear used as in test 6; one tooth broken or jambed. 80–T gear broken into 4 pieces, split at key-way. Key, obviously too high, canted. Ground off 1/32 in. of top. No teeth stripped
46	May 21	30	100	720	...	70	70	246	"German" key used. Same 30–T gear used as in test 6; not broken at all. 100–T gear broken into 6 pieces, split at key-way. No teeth stripped
47	May 21	40	40	720	...	30	40	114	Driver 40–T gear stripped all teeth. Driven 40–T gear stripped 12 teeth and broke 2. "German" keys used

TABLE 4 TESTS OF 10 DIAMETRAL PITCH, CAST-IRON, 14½-DEG. INVOLUTE GEARS

COMPUTED RESULTS OF SECOND SERIES OF RUNS

SPEED CORRECTIONS MADE FOR ERROR OF TACHOMETER. BRAKE ARM = 32 IN. ZERO BRAKE LOAD = 3.5 LB., BALANCED ON UPPER SCALE-BEAM

Test No.	Teeth in Gears				Correct r.p.m. Brake Gear	Pitch Circumference in Ft. Brake Gear	Net Brake Load in Lb.	Pitch Speed Brake Gear, Ft. per Min.	Equivalent Load at Teeth, in Lb.	Remarks
	Change Gears		Tested Gears							
	Sprocket 8 Pitch, Steel Driver	Intermediate 8 Pitch, Steel Driven	Intermediate 10 Pitch, Cast Iron Driver	Brake 10 Pitch, Cast Iron Driven						
1	20	120	20	30	24.6	0.7854	68	19.32	1451	Reliable run
2	20	120	20	30	24.7	0.7854	78	19.40	1664	Reliable run
3	20	120	20	30	24.7	0.7854	76	19.40	1621	Reliable run
4	70	70	20	30	143.96	0.7854	62	113.07	1323	Reliable run
5	70	70	20	30	145.5	0.7854	58	114.28	1237	Reliable run
6	70	70	20	30	145.5	0.7854	60	114.28	1280	Reliable run. No teeth broken out of either gear
8	100	40	20	30	534.0	0.7854	52	419.40	1109	Reliable run. Tests Nos. 8–47 inclusive made at higher voltage than tests Nos. 1–7 inclusive
9	100	40	20	30	555.0	0.7854	58	435.90	1237	Reliable run
10	100	40	20	30	565.0	0.7854	58	443.75	1237	Reliable run
11	120	40	20	30	660.0	0.7854	54	518.36	1152	Reliable run
12	120	40	20	30	660.0	0.7854	60	518.36	1280	Reliable run. No teeth broken out of either gear
13	120	40	20	30	660.0	0.7854	52	518.36	1109	Reliable run
14	120	30	20	30	931.2	0.7854	48	731.36	1024	Nothing broken. Runs at this speed stopped. Too fast for brake
7	100	40	20	30	395.4	0.7854	52	310.55	1109	Nothing broken. Motor stalled. New electrical connections made
15	70	40	20	30	410.0	0.7854	52	322.01	1109	Reliable run, but load was probably a little less than 1109 lb. See Log, Table 3
16	70	40	20	30	406.0	0.7854	54	318.87	1152	Reliable run

TABLE 4—*Continued*

Test No.	Teeth in Gears				Correct r.p.m. Brake Gear	Pitch Circumference in Ft. Brake Gear	Net Brake Load in Lb.	Pitch Speed Brake Gear, Ft. per Min	Equivalent Load at Teeth, in Lb.	Remarks
	Change Gears		Tested Gears							
	Sprocket 8 Pitch, Steel Driver	Intermediate 8 Pitch, Steel Driven	Intermediate 10 Pitch, Cast Iron Driver	Brake 10 Pitch, Cast Iron Driven						
17	70	40	20	30	410.0	0.7854	56	322.01	1195	Reliable run, but load was probably a little less than 1195 lb. See Log, Table 3
18	20	120	30	40	44.93	1.047	116	47.05	1856	Reliable run
19	20	120	30	40	46.24	1.047	120	48 42	1920	Reliable run
20	20	120	30	40	45.59	1.047	123	47.72	1968	Reliable run
21	70	100	30	40	192.76	1.047	112	201.86	1792	Reliable run
22	70	100	30	40	186.02	1 047	108	194.80	1728	Reliable run
23	70	100	30	40	194.78	1.047	106	203.97	1696	Reliable run
24	70	70	30	40	277.3	1.047	102	290.39	1632	Reliable run
25	70	70	30	40	277.3	1 047	102	290.39	1632	Reliable run
26	70	70	30	40	277.3	1 047	103	290.39	1648	Reliable run
27	70	40	30	40	.....	1.047	92	.....	....	Neither gear broken. Motor stalled. Circuit breaker reset for heavier current
28	70	40	30	40	478.53	1.047	98	501.12	1568	The three following runs indicate that this is probably faulty. Same gears used as in test No. 27
29	70	40	30	40	.....	1.047	96	....	....	Neither gear broken. Motor stalled. Circuit breaker reset for heavier current.
30	70	40	30	40	.....	1.047	100	.....	....	Neither gear broken. Motor stalled. Circuit breaker reset for heavier current.
31	70	40	30	40	458 48	1.047	102	480.12	1632	Neither gear broken. Motor stalled. Abandoned attempt to run at this speed Same gears used in tests 29, 30 and 31.

32	100	70	30	40	396.14	1.047	108	414.84	1728	Reliable run
33	100	70	30	40	396.14	1.047	104	414.84	1664	Reliable run
34	100	70	30	40	396.14	1.047	106	414.84	1696	Reliable run
35	70	70	30	60	184.86	1.571	154	290.38	1643	Reliable run
36	70	70	30	60	184.86	1.571	156	290.38	1664	Reliable run
37	70	70	30	60	184.86	1.571	150	290.38	1600	Reliable run
38	70	70	30	80	136.72	2.094	202	286.35	1616	Reliable run, except that bearing slipped allowing load to come on ends of teeth, hence 1616 lb. load is too low
39	70	70	30	80	139.61	2.094	204	292.40	1632	Reliable run
40	70	70	30	80	138.65	2.094	206	290.39	1648	Reliable run, but no teeth were stripped
45	70	70	30	80	138.65	2.094	198	290.39	1584	Unreliable run. Gear split. No teeth stripped. "German" key; see Log, Table 3
41	70	70	30	100	110.92	2.618	250	290.39	1600	Unreliable run. Gear split. No teeth broken. See log, Table 3
46	70	70	30	100	110.92	2.618	246	290.39	1574	Unreliable run. Gear split. No teeth broken. See Log, Table 3, "German" key
42	70	70	30	30	369.73	0.7854	72	290.39	1536	Reliable run
43	70	70	30	30	369.73	0.7854	70	290.39	1493	Reliable run
44	70	70	30	30	369.73	0.7854	68	290.39	1451	Reliable run
47	30	40	40	40	277.30	1.047	114	290.33	1824	Reliable run

of the equivalent breaking loads was taken for each set, and the curve drawn through these average points as near as might be.

14 In Fig. 4 the curve was extended by inference to the zero velocity line. It will be noted that this breaking load curve falls off with increase of speed up to a pitch velocity of something less than 300 ft. per min. and then apparently starts to

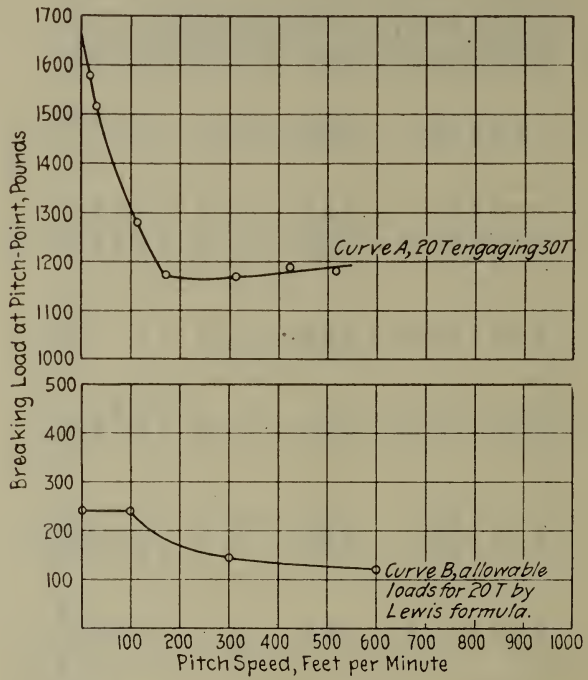


FIG. 4 RESULTS OF TESTS WITH 20-TOOTH PINIONS

rise again as though the maximum percussive effect had been passed. This fact, and the further one that within the limits of the actual tests the range of average breaking load was only from 1579 lb. at 19.4 ft. per min. down to 1169 lb. at 312.7 ft. pitch velocity, about 25 per cent, are two of the most striking points appearing in this investigation.

15 These phenomena might be questioned on the score that all of the 20-tooth gears but one broke entirely apart as well as having teeth stripped. But Fig. 5 shows the same general form for all tests involving 30- and 40-tooth gears in mesh;

and in this case there can be no question of anything but the strength of the teeth themselves, as it was only at the teeth that any of these gears broke.

16 It is an unfortunate fact that the limit of the motor's power was such as to make it impossible to rupture gears at higher speeds, thus enabling us to follow the curve and observe whether the apparent rise in breaking strength continued with increase in pitch speed. No attempt has been made to make this a smooth curve; the points corresponding to the averages obtained by experiment have merely been joined.

17 It is clear, however, even from these somewhat limited

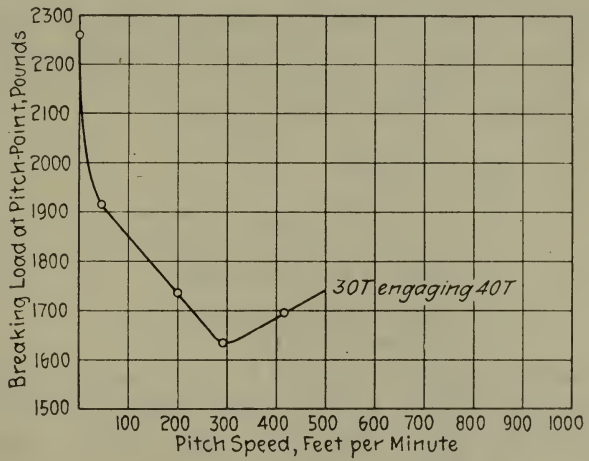


Fig. 5 Results of Tests with 30 and 40-Tooth Gears

experiments, that the impact or percussive effect with increase of speed is not nearly so great, with these modern, accurately cut gears, as has commonly been supposed. In both cases the curves show that the minimum breaking strength (at a pitch speed of about 300 ft. per minute) is more than seven-eighths that at a pitch speed of about 100 ft. per min. and more than seven-tenths that at zero pitch speed. And it is not theoretically impossible that the breaking load actually rises with increase of speed, after a certain critical speed has been passed, until it would approximate the static breaking load if it were not for the increasing tendency of the teeth to tear loose, due to centrif-

ugal force. At an infinite speed the repetitive stress would become a continuous one. As Curve *A*, Fig. 4, shows the actual breaking strength of 20-teeth, 10-pitch, cast-iron gears at different speeds, it is interesting to compare with this the allowable loads for such gears computed for the same speeds by the Lewis formula (see Appendix No. 1). Curve *B*, expressing these, is shown at the bottom of Fig. 4. The discrepancy between the two curves is obvious. They do not even have the same general form.

18 From Par. 4 it is seen that Mr. Lewis's formulae and tables were intended to provide a factor of safety of 3 for pitch speeds of 100 ft. per min. or less, of 5 for 300 ft., and 6 for 600 ft., the differences being intended to cover presumed increases in impact effect. Using Mr. Lewis's accepted formula for 15-deg. involute teeth as being applicable to the Brown & Sharpe standard 14½-deg. involute teeth, namely,

$$W = spf\,0.124 - \frac{0.684}{n}$$

W = working strength of tooth
s = working stress of material per sq. in.
p = pitch of teeth in inches
f = face of teeth in inches
n = number of teeth

Table 5 has been prepared to show the relation between the computed working strength and the actual breaking loads for the 20- and 30-tooth gears within the range of these experiments as shown by the plotted curves or their legitimate inferential extension.

19 When the actual breaking loads under running conditions have been determined, as they have in the above cases, the results provide the allowances for pitch-speed variations and it is difficult to see any reason for not using a uniform factor of safety for conditions which are uniform in other regards than pitch-speeds, to provide for the possibilities of faulty material, poor alignment, sudden applications or reversals of stress, and possible overloads. Just how great this factor need be is a question to be settled by the designer in each individual case. But in any case it would be a true factor of safety and not a factor of ignorance.

20 Mr. Lewis's Table II may be looked upon as a table of

TABLE 5 RELATION BETWEEN COMPUTED WORKING STRENGTH AND BREAKING LOADS 20- AND 30-TOOTH GEARS

20T In Mesh With 30T	Pitch Speed, Ft. per. Min. 0	100	300	600
Computed Working Load, Lewis's Formula	240	240	144	120
Actual Breaking Load, Tests	1660	1310	1165[1]	1165[1]
Corresponding Factor of Safety	6.9	5.5	8.1	9.7
Working Load, Factor of Safety =4	413	340	290	290
30T In Mesh With 40T				
Computed Working Load, Lewis's Formula	270	270	163.2	135
Actual Breaking Load, Tests	2260	1850	1640	1640[1]
Corresponding Factor of Safety	8.4	6.9	10	12.2
Working Factor of Safety =4	565	463	410	410

[1] Neglecting apparent rise in curve.

TABLE 6 VELOCITY COEFFICIENTS

Pitch Velocity, Ft. per Min.	000	100	150	200	300	400	500
Ratio $\frac{\text{Breaking Load at Given Vel.}}{\text{Breaking Load at Zero Vel.}}$ Data from Fig. 4	$\frac{1660}{1660}=1.0$	$\frac{1310}{1660}=0.789$	$\frac{1205}{1660}=0.725$	$\frac{1165}{1660}=0.702$	$\frac{1165}{1660}=0.702$	$\frac{1175}{1660}=0.707$	$\frac{1185}{1660}=0.712$
Ratio $\frac{\text{Breaking Load at Given Vel.}}{\text{Breaking Load at Zero Vel.}}$ Data from Fig. 5	$\frac{2260}{2260}=1.0$	$\frac{1850}{2260}=0.819$	$\frac{1790}{2260}=0.792$	$\frac{1735}{2260}=0.768$	$\frac{1640}{2260}=0.726$	$\frac{1680}{2260}=0.743$	$\frac{1740}{2260}=0.770$
v—Velocity Coefficient, Safe	1.00	0.80	0.75	0.72	0.70	0.68	0.66

factors representing a combination of the modulus of rupture, a factor of safety, and a coefficient depending upon velocity. It would seem better to separate these. The value of the modulus of rupture is discussed in Par. 22. For values to be taken for the factor of safety the writer agrees with Mr. Christie as quoted in Par. 4.

21 The values of the coefficient to provide for the effect of pitch-velocity v can be determined from the results of the tests shown in Fig. 4, Curve A, and Fig. 5. These results are expressed in Table 6, together with a series of values of v based upon them and so selected as to lean toward the side of extra safety.

22 To check the material of the gears used in these tests and to determine whether it was normal or of exceptional strength, 24 test specimens were cut from 12 of the 30-tooth gears, selected at random, as shown in Fig. 6. These specimens were about ¼ in. thick, 1 1/16 in. wide and 2¼ in. long. Seventeen similar specimens were cut from castings made in the University foundry and known to be of ordinary composition and quality. The specimens were all tested in flexure with the load applied at the middle of a span of 1¾ in. The data and results of these tests are given in full in Table 7. The modulus of rupture of the gear material test pieces was 38,737 lb., while that of the check specimens was 39,049 lb. It is legitimate to conclude that the material of the gears used in these tests, while of first-rate quality, was not unusual or exceptional. The results of the tests may therefore be accepted as representing typical material. In general a value of 36,000 may be reasonably used for the modulus of rupture for cast-iron gear teeth.

23 The Lewis formula is based upon the theory of flexure under the assumption that the entire load is borne by a single tooth, the point of application of the load being the extreme end of the tooth, and the direction of the acting force being normal to the tooth profile at that point. The line of action of the force is considered carried to the center line of the tooth (Fig. 7) with W equal to the transverse component as applied at this point. Subsequently it treats W as the load at the pitch circumference. The effects of the compressive and shear components are ignored. Fracture is assumed to take place on a plane section, the position of which is determined by inscribing a parabola of uniform

strength with the vertex lying at the intersection of the center line of the tooth and the line of action of the force. The dangerous section is the plane section containing the points of tangency of the parabola with the tooth outline. That these assumptions are too severe or unfavorable for accurately cut gears is evident from the result of these tests, as will be shown later.

24 To check the applicability of the Lewis formula to the

FIG. 6 METHOD OF OBTAINING TEST SPECIMENS

form of gear tooth employed in these experiments, the teeth of the 20, 30, 40, 60, 80, 100 and 150 tooth 14½-deg. involute gears were carefully measured by gear-tooth micrometer calipers. They then, together with a theoretical rack tooth of the same system, were accurately drawn to a scale of 100 times full size. In these teeth, parabolas were inscribed by the Lewis method and the dangerous section located. The data and results are given in Appendix No. 2. The factors for strength do not seriously vary from those given by the Lewis expression[1] for 15-deg involute teeth, i.e., $0.124 - \frac{0.684}{n}$. If the assumptions of Par. 23 are

[1]American Machinist, June 22, 1893.

TABLE 7 TESTS OF MATERIALS, DATA AND RESULTS, JUNE 26, 1912

SPECIMENS CUT FROM GEARS

Mark	Thickness	Width	Breaking Load at Mid-point	Corresponding Modulus of Rupture	Average Modulus of Rupture	Remarks
05*A*	0.251	1.066	980	38305	.....	
05*B*	0.251	1.065	975	38145	38225	
08*A*	0.251	1.065	1000	39123	.....	
08*B*	0.251	1.063	1000	39196	39160	Small blow-hole on top.
09*A*	0.248	1.066	1167.5	46343	.....	
09*B*	0.250	1.065	1197.5	47225	46784	
1*A*	0.251	1.063	917.5	35969	.....	Small blow-hole on bottom.
1*B*	0.251	1.063	967.5	37922	36946	
17*A*	0.248	1.060	1106	44533	.....	
17*B*	0.250	1.058	1156.5	45910	45222	
19*A*	0.245	1.060	950	39194	.....	
19*B*	0.248	1.061	970	39020	39107	
28*A*	0.251	1.067	.....	.....	.....	Flawed. No reading.
28*B*	0.251	1.067	850	33192	33192	
32*A*	0.251	1.060	1000	39307	.....	
32*B*	0.250	1.061	995	39387	39347	
37*A*	0.251	1.063	895	35081	.....	
37*B*	0.251	1.063	904	35434	35258	
39*A*	0.251	1.0635	842.5	33008	.....	
39*B*	0.250	1.0635	870	34358	33683	
43.*A*	0.251	1.060	867.5	34099	.....	
43.*B*	0.251	1.060	874	34355	34227	
43..*A*	0.251	1.056	1087.5	42909	.....	
43..*B*	0.251	1.054	1125	44472	43691	
				Grand Aver.	38737	

CHECK SPECIMENS, ORDINARY CAST IRON

A	0.251	1.0615	1000	39252	.....	
B	0.251	1.063	.....	.....	.....	Broke before initial load, 900 lb.
C	0.250	1.058	990	39300	.....	
D	0.252	1.062	962.5	37463	.....	
E	0.252	1.062	975	37862	.....	
F	0.251	1.059	1000	39354	.....	
G	0.252	1.062	920	35809	.....	
H	0.248	1.059	990	39899	.....	
I	0.257	1.062	1195	44720	.....	
J	0.254	1.058	880	33842	.....	
K	0.251	1.057	1018	40129	.....	
L	0.247	1.061	842	34145	.....	
M	0.245	1.064	.....	.....	.....	Void. Flawed.
N	0.250	1.0625	1014	40083	.....	
O	0.250	1.0625	1136	44905	.....	
P	0.252	1.058	945	36930	.....	
Q	0.252	1.062	1080	42037	.....	
				Grand Aver.	39049	

Distance between supports = 1.75 in. in all cases.

correct, the Lewis formula should give the actual static breaking loads when the actual modulus of rupture is inserted for s.

25 To test this, consider first the case of a 20-tooth 10-pitch, 1 1/16-in. face, cast-iron pinion, the material having a modulus of rupture of 39,000. Substitution in the Lewis formula

$$W = spf\left(0.124 - \frac{0.684}{n}\right)$$

gives

$$W = 39{,}000 \times 0.31416 \times 1.0625\left(0.124 - \frac{0.684}{20}\right)$$

Therefore

$$W = 1170 \text{ lb.}$$

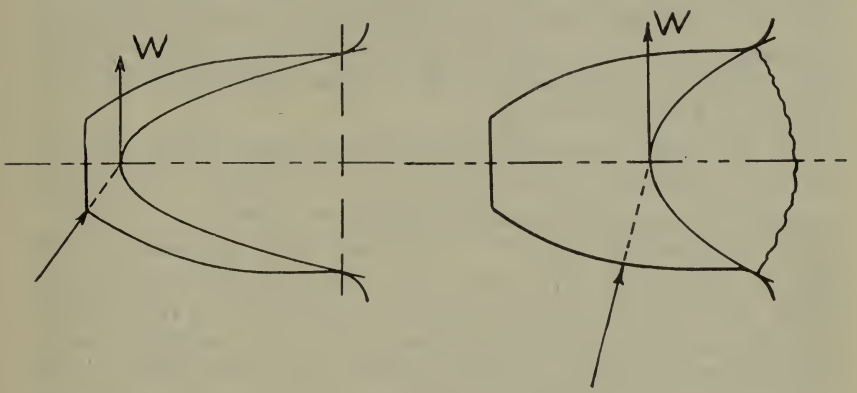


FIG. 7 LINE OF FORCE AS DETERMINED IN LEWIS FORMULA

FIG. 8 LINE OF FORCE AS DETERMINED BY ACTUAL FRACTURES

But from Curve A, Fig. 4, it is seen that the static strength (i.e. equivalent breaking load at zero pitch velocity) is approximately 1650 lb. for a 20-tooth pinion in mesh with a 30-tooth gear. The value computed by the Lewis formula, when using the real modulus of rupture, is only two-thirds of the actual value shown by the tests. Similarly, the Lewis formula gives a value of 1317 lb. for the static breaking strength of the 30-tooth gear, while Fig. 5 shows this to be, for a 30-tooth meshing with a 40-tooth, 2260 lb. Again the Lewis value is much too low, being but 60 per cent of the actual test result.

26 These discrepancies demonstrate that the assumptions

upon which the Lewis formula are based are too severe and they indicate the necessity of taking into consideration the length of the arc of action; for if the length of the arc of action exceeds the pitch arc, it is evident that the previously engaging teeth will not have gone out of action at the time when the load first comes on the newly engaging teeth, and it is only at the instant of engagement or disengagement that the load can be applied at the end of a tooth. This is not only theoretically so but it can be counted on with reasonable certainty for modern, accurately cut gears. It must be borne in mind that this is the least stiff position of the tooth and that it would therefore tend to yield slightly under the load, thus allowing another pair of teeth to come into engagement or remain in engagement. This elastic yielding would be apt to make up for the slight errors in tooth spacing. The mathematical discussion of the arc of action and its numerical determination for the gear combinations used in these experiments will be found in Appendix No. 3.

27 In order to test the actual static strength of the teeth in various positions, and with one, two or if possible, three pairs engaging simultaneously, experiments were carried out with results as shown in Table 8. The loads were applied in each case by a steel 30-tooth pinion with all teeth save three single ones, two pairs of two, and one set of three having been milled off in such a way as to leave gaps between the remaining teeth. Because of this there could be positive assurance that only as many pairs of teeth engaged in any test as it was intended. The gears were set up with the teeth engaging in the position desired, the cast-iron gear to be tested being the one on the brake shaft and a surface gage being used to test the accuracy of conditions. All play was taken up and the brake was tightened down with emery cloth between it and the wheel so as to prevent all slip. The breaking power was supplied by means of a special hand-operated lever instead of by motor.

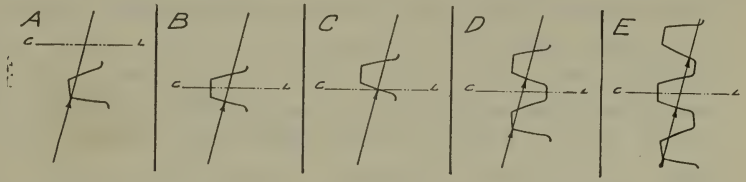
28 Position *A* corresponds to the loading assumptions of the Lewis formula and it is interesting to compare these results with those obtained by using the Lewis formula with a modulus of rupture of 39,000 lb. Table 9 presents this comparison in concise form.

TABLE 8 EQUIVALENT STATIC BREAKING LOADS W AT PITCH LINE—SERIES 4, SEPTEMBER 11-13, 1912

10-Pitch, Cast-Iron Gear

Position of Stressed Teeth and Equivalent Breaking Load

Single Tooth Loaded | 2-Teeth Loaded | 3-[?] Teeth Loaded



	Test No.	W	Test No.	W	Test No.	W	Test No.	W	Test No.	W
20 T	4	1216	12	1440	1	1632	7	2112	9	1600
	5	1184	13	1476	2	1696	8	2176	10	1472
	6	1152	..	....	3	1664	..	....	11	1632
	Aver.	1184	Aver.	1458	Aver.	1664	Aver.	2144	Aver.	1568
30 T	18	1472	25	1877	14	1941	20	2603	23	1835
	19	1429	26	1707	15-16	Void	21	2176	24	1856
	..	....	27	1728	17	1899	22	2965	..	....
	Aver.	1451	Aver.	1771	Aver.	1920	Aver.	2581	Aver.	1846
40 T	31	Void	39	2416	28	2416	34	3360	36	Void
	32	1616	40	2336	29	2272	35	3360	37	2560
	33	1600	..	....	30	2208	..	....	38	2528
	Aver.	1608	Aver.	2376	Aver.	2299	Aver.	3360	Aver.	2544

TABLE 9 EQUIVALENT BREAKING LOADS

Size of Gear	20 T	30 T	40 T
W by Test, Position A	1184	1451	1608
W computed by Lewis Formula	1170	1317	1392

29 The test values of W, as given in Table 9, correspond to a formula for this position of engagement and static conditions:

$$W = spf\left(0.154 - \frac{1.26}{n}\right)$$

The derivation of this formula from the experimental results will be found in Appendix No. 4. This equation would make the static strength of single teeth vary from that of a theoretically correct rack tooth down to a 20-tooth pinion in the ratio of 0.154 to 0.091; in other words, the rack tooth would have 1.69 times the strength of the 20-tooth pinion. The Lewis formula would give a ratio of 0.124 to 0.090 for the same gears. The discrepancy can be explained by the fact that the theory of flexure is recognized as only approximately correct, particularly for oblique forces, material which has not the same strength in tension as in compression, and for stresses beyond the elastic limit. Furthermore the Lewis factors neglect entirely the effect

of the angularity of the force in producing a compressive stress. In the case of the 20-tooth gear this angularity is such as to make the compressive component of the acting force more than 50 per cent of the transverse component. The theory of flexure upon which the Lewis formula is based would have the fracture occur on a plane section (as shown in Fig. 7) and would have the breaking loads vary, approximately as the square of the thickness of the tooth at this section. As a matter of fact the fracture of the teeth in every case took place lower down on the root than at the points of tangency of parabolas inscribed on the Lewis assumption. The fractures took place rather at the points of tangency of parabolas constructed for loads applied at the pitch points as shown in Fig. 8, and the breaking loads varied more nearly in proportion to the squares of the thickness at this section. The surface of the break invariably was in the form of a curve running down into the root as shown and in no case did a tooth break square across.

30 Mention is made in Par. 26 of the necessity of taking the arc of action into consideration in dealing with the load-carrying strength of gear teeth. It would seem that the breaking load of a gear would depend not solely upon itself but also upon the gear with which it meshed. The larger the meshing gear, the greater the arc of action; and the greater the arc of action, the more likely the load is to be borne on two or more pairs of teeth simultaneously and the less likely that the entire load will fall on a single tooth in its position of least strength. It is desirable to see, then, what effect, if any, the length of arc of action has on the breaking load. To see what the effect was under static (zero pitch velocity) conditions, a series of tests (Series 5) was conducted. Tests were made of 30-tooth gears in mesh with 30, 40, 60 and 80 tooth gears. It was found in making these tests that the position of the teeth in engagement made a vast difference in results. It was therefore necessary to find by trial the position of engagement corresponding to the least carrying power of the teeth. The results are given in Table 10. There is nothing to account for the foregoing variation in strength except the differences in arc of action. It was the intention to carry similar tests out on 30-tooth meshing with 80-tooth and 100-tooth, but it was found in trying the 30 and 80 pair, that the rim of the 80-tooth broke, cracking the arms and hub, before a breaking load of the teeth could be reached. Fig. 9,

TABLE 10 STATIC STRENGTH. SERIES 5, SEPTEMBER 26–27, 1912

Breaking Loads W of 30T in mesh with

30T			40T			60T		
Series	Test	W	Series	Test	W	Series	Test	W
4	23	1835	5	1	2480	5	14	2453
4	24	1856	5	6	2368	5	15	2453
5	17	2475	5	7	2080			
5	18	2304	5	8	2112			
Average.....2118			Average.....2260			Average.....2453		
Arc of Action, In...0.61158			Arc of Action, In...0.62814			Arc of Action, In...0.64919		

Curve A, shows the results of these tests, the ordinates being equivalent breaking loads at pitch point, and the abscissae being arcs of contact. There being no unbroken 20-tooth gears to use in Series 5, the value for static strength of 30-tooth meshing with 20-teeth (arc of action, 0.53938 in.) is taken from Fig. 4, Curve A.

31 From Table 9 it is seen that the static strength of a single tooth of a 30-tooth gear in its weakest position = 1451 lb. This would measure the strength of teeth with an arc of action of 0.314 in. or less, i.e. when the $\frac{\text{arc of action}}{\text{pitch arc}} \leq 1$. From Fig. 9, Curve A, for a 30-tooth meshing with a 20-tooth the arc of action equals 0.53938 in. and the corresponding $W = 1660$ lb. From these facts and Table 10 it is possible to construct Table 11 to show the relationship, under static conditions, existing between length of arc of action and equivalent breaking load.

TABLE 11 RELATION BETWEEN ARC OF ACTION AND EQUIVALENT STATIC LOAD

Ratio $\frac{\text{Arc of Action}}{\text{Pitch Arc}}$	$\frac{0.31416}{0.31416}=1$	$\frac{0.53938}{0.31416}=1.72$	$\frac{0.61158}{0.31416}=1.95$	$\frac{0.62814}{0.31416}=2$	$\frac{0.64919}{0.31416}=2.07$
Breaking Load.....	1451	1660	2118	2260	2453
Ratio $\frac{\text{Breaking Load}}{1451}$	1	1.14	1.46	1.56	1.69

The range in breaking strength due to varying arcs of action, from 1 to 1.69 shown by this table, is as great as that due to *form* between 20 teeth and 150 teeth (see Par. 29); and it is greater than that due to variations in velocity from zero pitch speed to over 500 ft. per min. (see Par. 21). If form of tooth and velocity are of sufficient importance to be taken into account in a formula for gear tooth strength it is demonstrated that there should also be introduced a factor for arc of action.

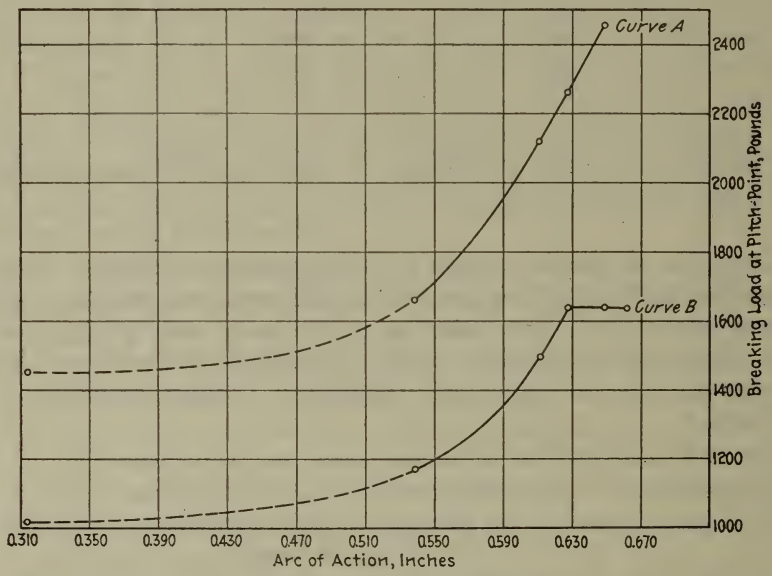


FIG. 9 RELATION BETWEEN EQUIVALENT BREAKING LOAD AND ARC OF CONTACT

32 Further light is thrown upon the extent of the influence of the length of arc of action if from Series 1 and 2 the tests involving 30-tooth gears made at a nearly uniform pitch velocity are selected. The following table gives the results of such tests made at about 290-ft. pitch velocity. Such a selection makes the arc of action the only variable factor. The values of Table 12 are shown graphically in Fig. 9, Curve *B*. There are no running experiments involving an arc of action of 0.31416. But from Figs. 4 and 5 it is clear that the breaking strength at 300 ft. per min. is about 70 per cent of the static strength. It is therefore not improper to assume that the break-

TABLE 12 RELATION BETWEEN ARC OF CONTACT AND BREAKING LOAD

Run	Speed	Gears	Arc of Contact	Breaking Load
06	290.03	30–20	0.53938	1280
7	310.55	30–20	0.53938	1109
15	322.01	30–20	0.53938	1109
16	318.87	30–20	0.53938	1152
17	322.01	30–20	0.53938	1195
				Average....1169
42	290.39	30–30	0.61158	1536
43	290.39	30–30	0.61158	1493
44	290.39	30–30	0.61158	1451
				Average.....1493
24	290.39	30–40	0.62814	1632
25	290.39	30–40	0.62814	1632
26	290.39	30–40	0.62814	1648
				Average....1637
35	290.38	30–60	0.64919	1643
36	290.38	30–60	0.64919	1664
37	290.38	30–60	0.64919	1600
				Average....1636
38	286.35	30–80	0.66209	1616[1]
39	292.40	30–80	0.66209	1632
40	290.39	30–80	0.66209	1648[1]
				Average....1632

[1] Data indicate that these values were lower than real strength of teeth. See Table 3.

ing strength for an arc of contract of 0.31416, at a pitch velocity of 300 ft. per min., would be 70 per cent of 1451 lb., or 1016 lb. On this assumption and the data of Table 11, Table 13 has been computed.

It is interesting to note how closely this agrees with Table 9.

33 The results in Tables 11 and 13 may now be combined and a table of coefficients a constructed by which the effect of the arc of action may be introduced into the formula for the strength.

TABLE 13 RELATION BETWEEN ARC OF ACTION AND BREAKING LOAD UNDER RUNNING CONDITIONS

Ratio: $\frac{\text{Arc of Action}}{\text{Pitch Arc}}$	$\frac{0.31416}{0.31416}=1$	$\frac{0.53935}{0.31416}=1.72$	$\frac{0.61158}{0.31416}=1.95$	$\frac{0.62814}{0.31416}=2$	$\frac{0.64919}{0.31416}=2.07$	$\frac{0.66209}{0.31416}=2.11$
Breaking Load	1016	1169	1493	1637	1636	1632
Ratio: $\frac{\text{Breaking Load}}{1016}$	1	1.15	1.47	1.61	1.61	1.61

TABLE 14 COEFFICIENTS BASED ON ARC OF ACTION

Ratio: $\frac{\text{Arc of Action}}{\text{Pitch Arc}}$	1	1.4	1.6	1.7	1.8	1.9	1.95	2.00
Corresponding a	1	1.05	1.1	1.15	1.24	1.38	1.47	1.60

34 The entire formula as derived from these experiments for the safe equivalent load at pitch line may now be written

$$W=\frac{spf}{k}\left(0.154-\frac{1.26}{n}\right)va$$

in which

W = safe working load at pitch line in pounds
s = modulus of rupture = 39,000 in these tests but ordinarily to be taken = 36,000
p = circular pitch in inches
f = width of face in inches
k = factor of safety
n = number of teeth in gear
v = velocity coefficient from Table 6
a = arc of action coefficient from Table 14

35 The general effect of arcs of action and of pitch speed on the breaking strength of gear teeth as found in these experiments can best be seen by combining Fig. 4, Curve *A*, Fig. 5 and

Fig. 9, Curves A and B into a single isometric chart. This is done in Fig. 10. The ordinates, as heretofore, represent equiva-

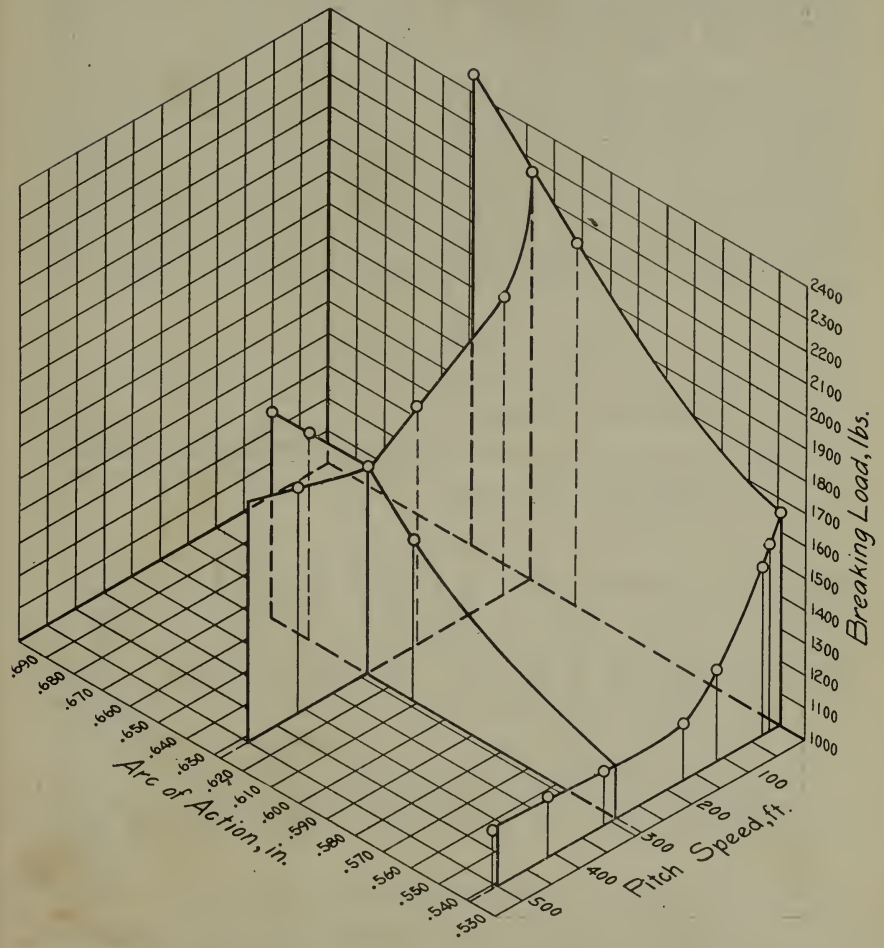


Fig. 10 Relations Between Arc of Action, Pitch Speed and Breaking Load Shown Isometrically

lent breaking loads at pitch point in pounds. One set of abscissae represents pitch speeds in feet per minute; the other set represents lengths of arc of action in inches.

36 To complete the record of this experimentation mention should be made of the tests which were designated as Series 3. In planning these it was thought that the remaining unbroken gears could have their width of face turned down to one-half its previous value, and then, the teeth being theoretically just half as strong as those of the full-width gears, tests might be conducted at higher pitch speeds and the gears ruptured within the capacity of the motor. The observed breaking load could then be doubled as the breaking load of the full-width gears and points so obtained be used to extend the curve of direct observations. The results of a very few tests proved that this could not be done. The data and results are briefly tabulated in Table 15.

TABLE 15 SERIES 3, JULY 1, 1912

Test No.	Width of Face	Change Gears		Test Gears		Motor R.P.M.	Net Brake Load, Lb.	PitchSpeed Brake Gear Ft.perMin.	Equivalent Load at Teeth, Lb.
		Sprocket	Intermediate	Intermediate	Brake				
48	17/32	70	40	30	30	720	28	508.18	597
49	1 1/16	70	40	30	30	720	66	508.18	1408
50	17/32	40	20	30	30	720	28	580.77	597
51	1 1/16	40	20	30	30	720	62	580.77	Could not break. Circuit-Breaker out.
52	1 1/16	70	40	30	30	720	66	508.18	1408 Same Gears as Test 51

It is evident from Tests 48-49 and 50-51 that the half-width teeth have not one-half the strength of the full-width teeth. Only speculations can be offered as to this apparent discrepancy. It may be that the wider teeth forced better alignment conditions without stressing the teeth unduly; or it may be that the additional mass plays some part in shock absorption. But since observations could not be used to supplement those on full-width gears, no further runs were made.

37 It would appear that the factors of safety in common use,

in so far as breaking strength of the teeth is concerned, while they might properly serve for rough cast gears, are much too large for modern, accurately cut gears; and that Mr. Lewis and Mr. Christie were quite right in assuming that the factor of safety at any speed need not be more than double that which will suffice for the slowest speed.

38 A few words may be added about the general design of the gears. The almost invariable breaking of the 20-tooth gears, 1 1/16-in. bore, ¼-in. keyway, tends to indicate that this is just about the maximum limit of bore which can be used with these gears while getting the maximum out of the tooth strength. Any larger bore, it seems, would simply allow these gears to split at the keyway without stripping any teeth. Of the 30- and 40-tooth gears, and of the proportions of the 60-tooth there can be no criticism. In all cases the failure came at the teeth, although it is indicated that the 60-tooth gears could have borne heavier loads if they had been solid. As a whole the tests involving gears with more than 60 teeth were indecisive so far as throwing any light on the strength of the teeth was concerned. These gears were neither solid nor webbed but had arms, and their less rigid design allowed them to yield and fracture at rim, arms and hub before the breaking strength of the teeth could be reached. To bring the rest of the gear up to the strength of the teeth would seem to require a larger diameter of hub, thicker rims and heavier arms. In loading the gears up to the breaking strength of the teeth considerable difficulty was encountered with shearing and battering of keys. This was practically overcome by the use of hardened tool-steel keys. A photographic log (Figs. 11-28) follows which shows the manner in which the gears failed.

ACKNOWLEDGMENT

The writer wishes to acknowledge his appreciation of the many kindnesses of his colleagues and the laboratory assistants. Particular thanks are due to Mr. L. E. Cutter, instructor in mechanical engineering, without whose unfailing assistance the tests could scarcely have been completed. All observations were made by the writer and Mr. Cutter personally.

FIG. 11 KEY SHEARED IN TEST 01, SERIES 1

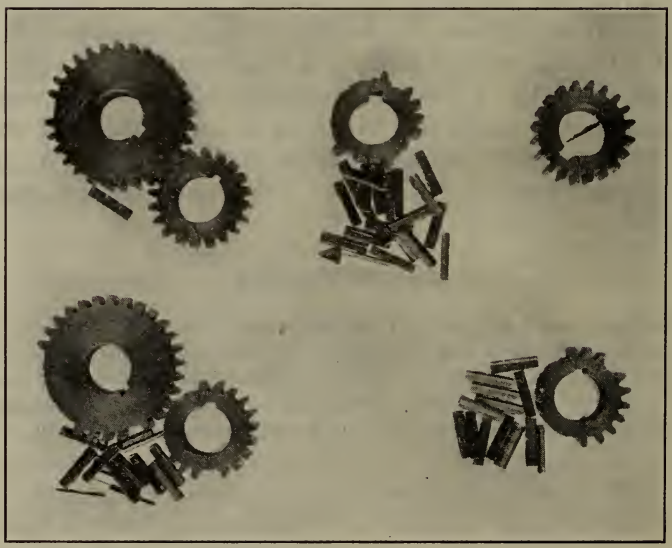


FIG. 12 20 TOOTH ENGAGING 30 TOOTH, TESTS 04-08 INCLUSIVE, SERIES 1

FIG. 13 20 TOOTH ENGAGING 30 TOOTH, TESTS 1-17 INCLUSIVE, SERIES 2

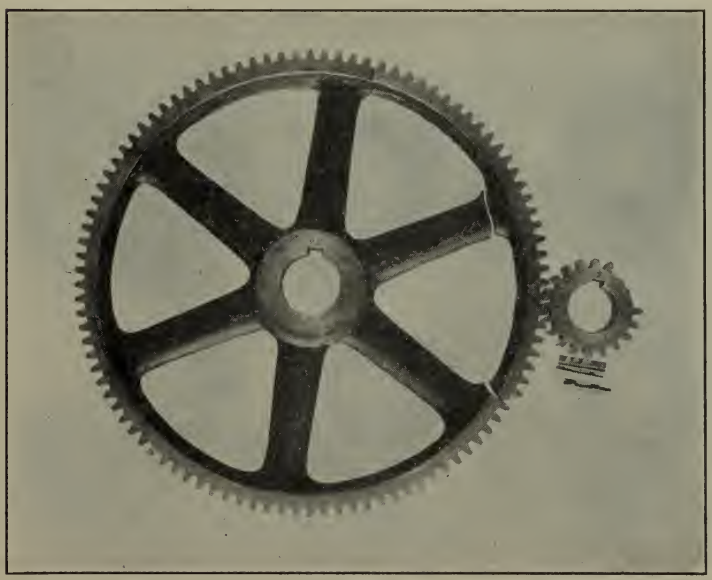


FIG. 14 20 TOOTH ENGAGING 100 TOOTH, TEST 03, SERIES 1

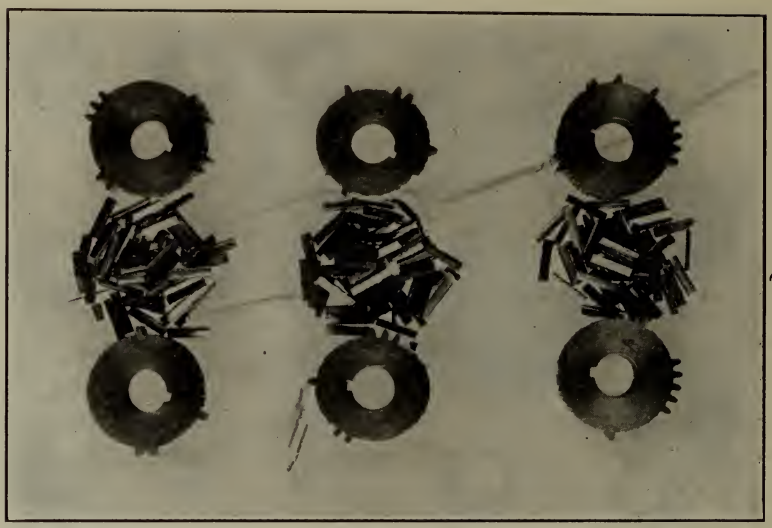


FIG. 15 30 TOOTH ENGAGING 30 TOOTH, TESTS 42-44 INCLUSIVE, SERIES 2

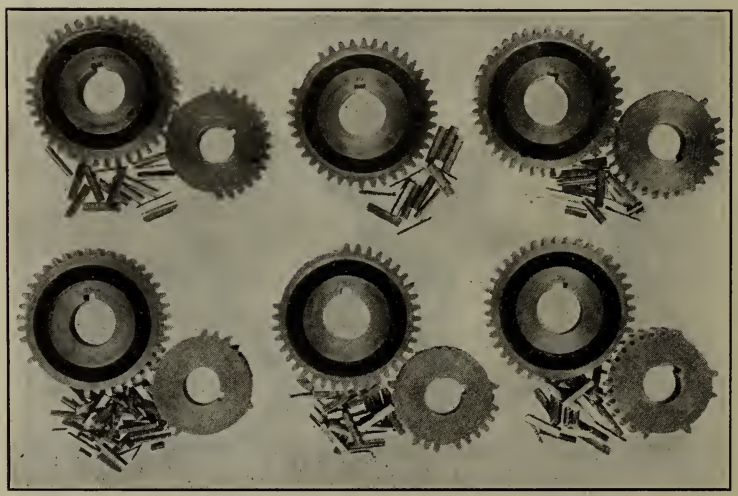


FIG. 16 30 TOOTH ENGAGING 40 TOOTH, TESTS 18-23 INCLUSIVE, SERIES 2

FIG. 17 30 TOOTH ENGAGING 40 TOOTH, TESTS 24-26, 32-34 INCLUSIVE, SERIES 2

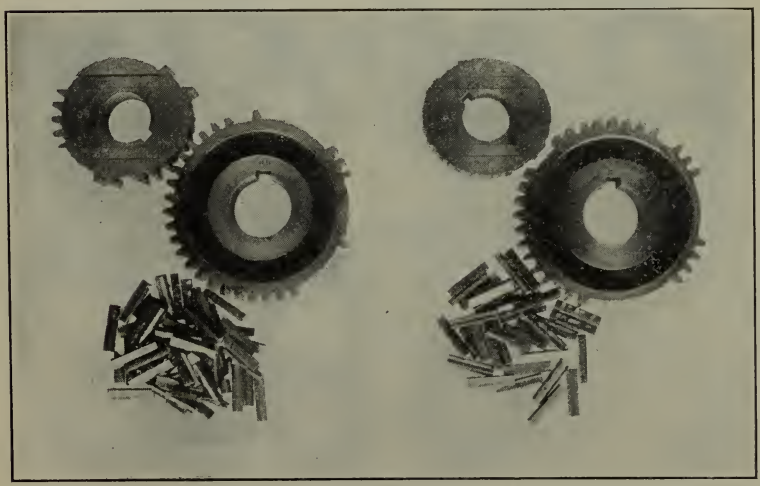


FIG. 18 30 TOOTH ENGAGING 40 TOOTH, TEST 09, SERIES 1 AND TEST 28, SERIES 2

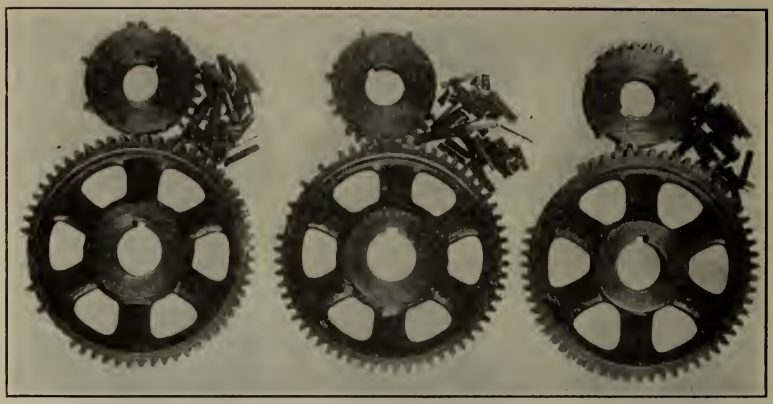


FIG. 19 30 TOOTH ENGAGING 60 TOOTH, TESTS 35-37 INCLUSIVE, SERIES 2

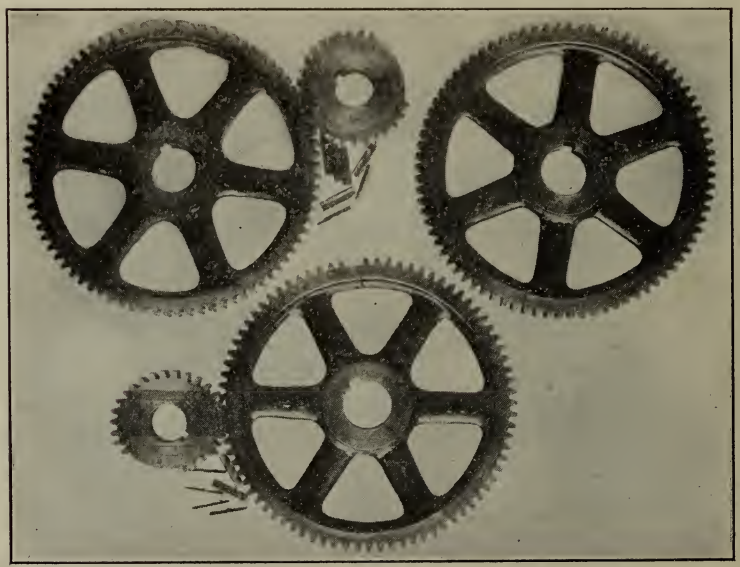


FIG. 20 30 TOOTH ENGAGING 80 TOOTH, TESTS 38-40 INCLUSIVE, SERIES 2

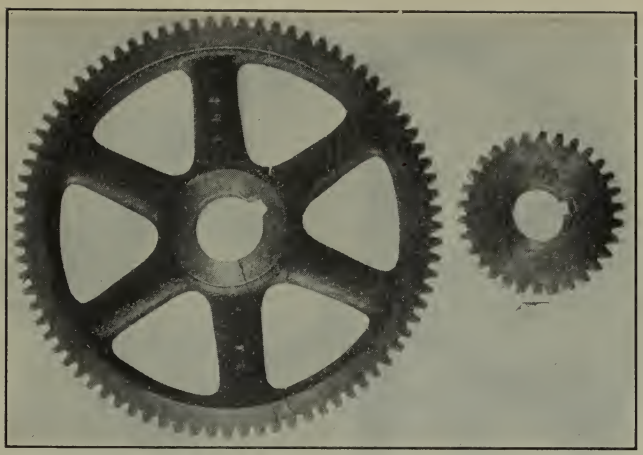


FIG. 21 30 TOOTH ENGAGING 80 TOOTH, TEST 45, SERIES 2. GERMAN HEXAGONAL KEY USED IN 80 TOOTH

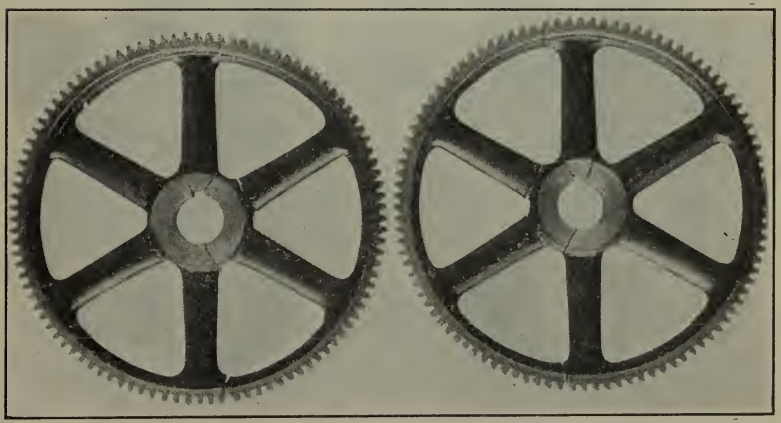


FIG. 22 30 TOOTH ENGAGING 100 TOOTH, TESTS 41 AND 46, SERIES 2

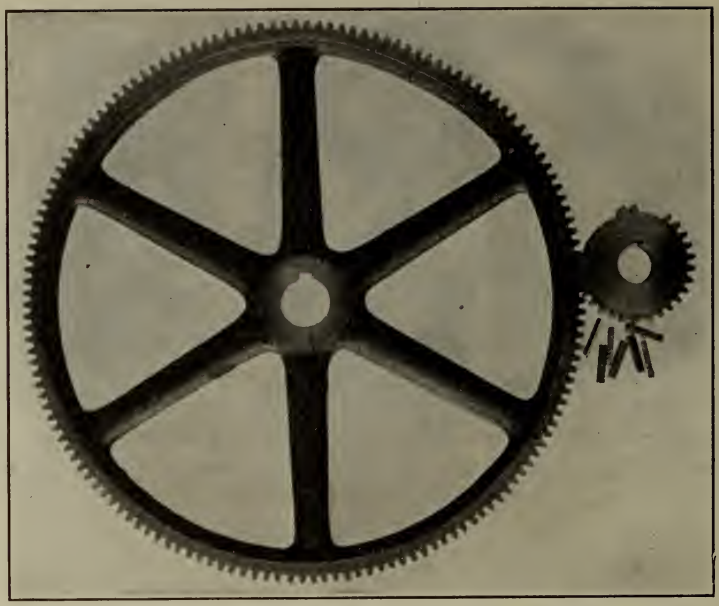


FIG. 23 30 TOOTH ENGAGING 150 TOOTH, TEST 016, SERIES 1

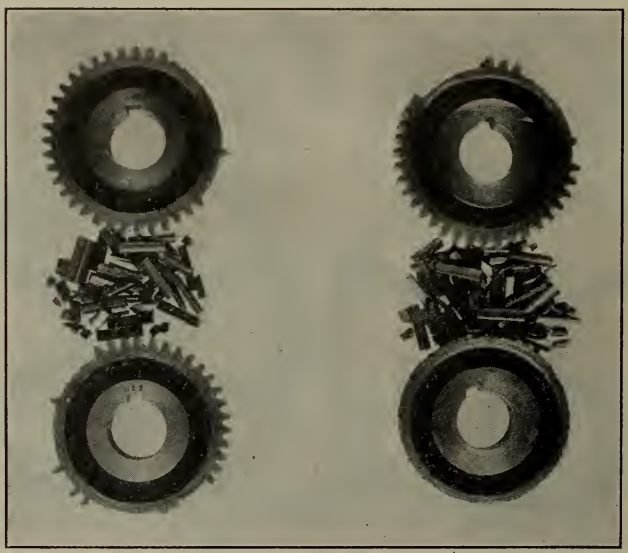


FIG. 24 40 TOOTH ENGAGING 40 TOOTH, TEST 011, SERIES 1 AND 47, SERIES 2

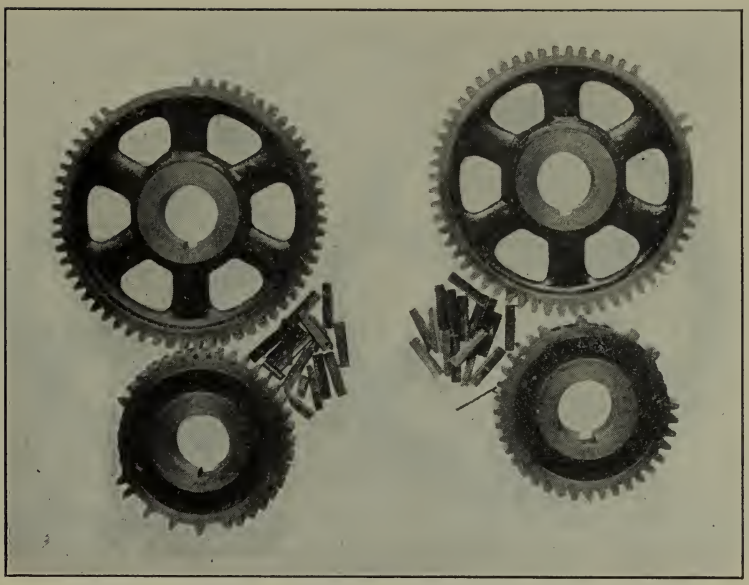


FIG. 25 40 TOOTH ENGAGING 60 TOOTH, TESTS 010 AND 012, SERIES 1

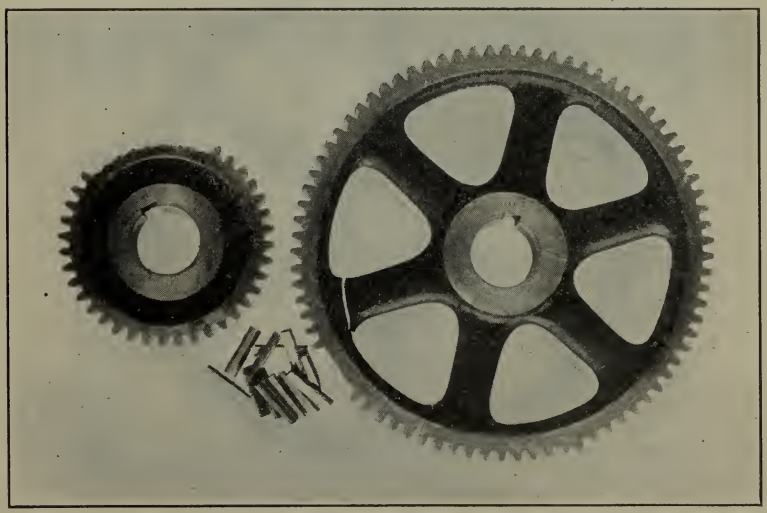


FIG. 26 40 TOOTH ENGAGING 80 TOOTH, TEST 013, SERIES 1

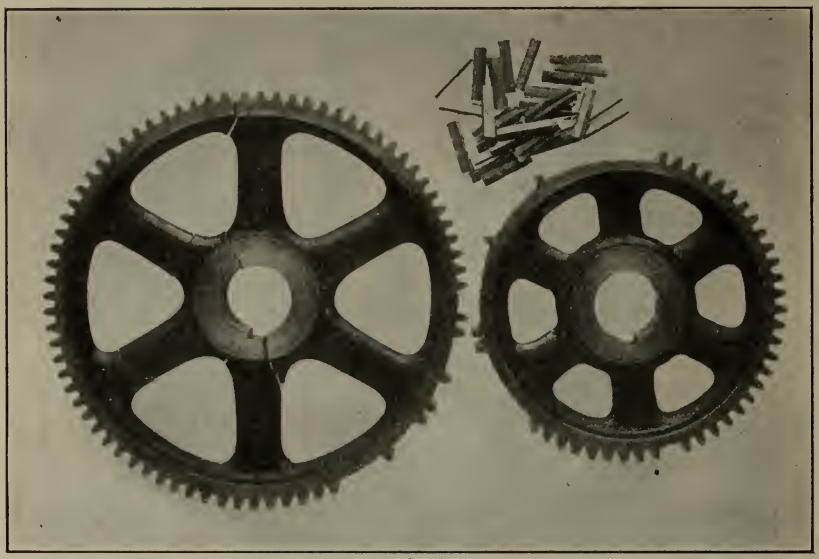


FIG. 27 60 TOOTH ENGAGING 80 TOOTH, TEST 014, SERIES 1

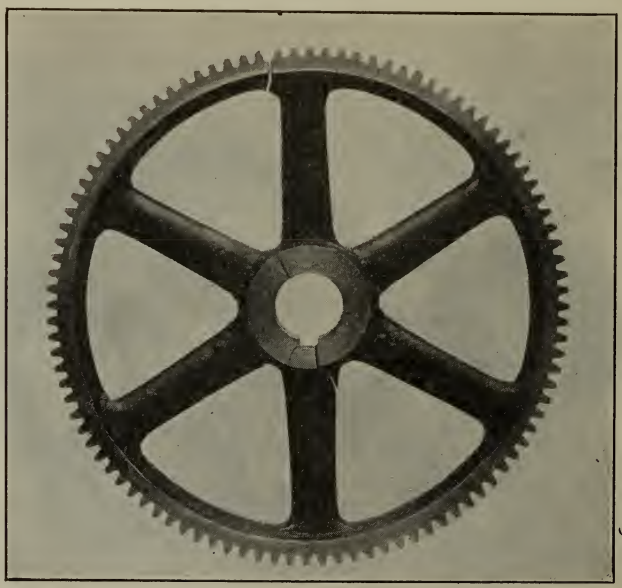


FIG. 28 80 TOOTH ENGAGING 100 TOOTH, TEST 015, SERIES 1. 80 TOOTH INTACT

APPENDIX No. 1

POINTS FOR FIG. 4, CURVE *B*

39 By Lewis formula

$$W = spf\left(0.124 - \frac{0.684}{n}\right)$$

where

$s = 8000$ for 0 to 100 ft. per min.

$W = 8000 \times 0.31416 \times 1.0625 \times 0.90 = 240$.....[1]

$s = 4800$ for 300 ft. per min.

$W = 4800 \times 0.31416 \times 1.0625 \times 0.90 = 144$.....[2]

$s = 4000$ for 600 ft. per min.

$W = 4000 \times 0.31416 \times 1.0625 \times 0.90 = 120$.....[3]

APPENDIX No. 2

40 To see how closely the factor of strength y obtained by the Lewis formula, $y = 0.124 - \frac{0.684}{n}$, corresponded to the forms of teeth used in this system, the following investigations were made:

41 The teeth were measured as accurately as possible, laid out 100 times full size, parabolas inscribed as shown in Figs. 7 and 8, and the distances

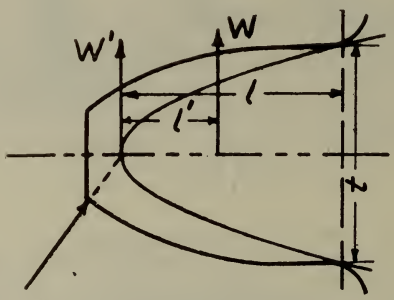


FIG. 29 SHOWING ASSUMED POINT OF APPLICATION OF FORCE AND RESISTANCE OF TOOTH

from vertices to pitch point, and from tangent lines to pitch point, measured. The results are given in Table 16.

42 Referring to Fig. 29, the load is assumed to be applied at the end of the tooth and normal to the profile. The transverse component $= W'$, and the equivalent force at the pitch line $= W$. The thickness of tooth at point of tangency of the inscribed parabola $= t$. The distance from the pitch line to vortex of parabola $= l'$, and from the vortex to the tangent line $= l$.

s = modulus of rupture

f = width of face in inches

$\frac{I}{c}$ = modulus of section

r = pitch radius of gear

Then, from the theory of flexure,

$$W'l = \frac{s\,I}{c} = \frac{sft^2}{6} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [4]$$

TABLE 16 DIMENSIONS OF GEAR TEETH

No. of Teeth	Thickness at Depth							End Load Parabola			Pitch Load Parabola		
	0.00	0.048	0.100	0.150	0.175	0.185	0.200	Thickness at Tangency	Tangent Line to Pitch Line	Vertex to Pitch Line	Thickness at Tangency	Vertex to Pitch Line	Tangent Line to Pitch Line
20	0.070	0.1240	0.1570	0.1630	0.1630	0.1630	0.1680	0.1662	0.0964	0.0760	0.1680	0.0152	0.1002
30	0.080	0.1277	0.1572	0.1678	0.1701	0.1710	0.1760	0.1750	0.0975	0.0740	0.1784	0.0150	0.1014
40	0.086	0.1260	0.1570	0.1730	0.1760	0.1785	0.1830	0.1820	0.0964	0.0797	0.1860	0.0160	0.1013
60	0.092	0.1300	0.1570	0.1745	0.1800	0.1840	0.1910	0.1896	0.0972	0.0800	0.1918	0.0158	0.1002
80	0.086	0.1260	0.1550	0.1760	0.1815	0.1850	0.1910	0.1902	0.0976	0.0789	0.1922	0.0178	0.1005
100	0.088	0.1240	0.1550	0.1750	0.1820	0.1870	0.1950	0.1920	0.0946	0.0815	0.1966	0.0178	0.1015
150	0.096	0.1300	0.1570	0.1800	0.1900	0.1930	0.2000	0.1980	0.0968	0.0815	0.2020	0.0191	0.1020
Rack	0.1053	.....	0.1570	.....	.....	.....	0.2094	0.2080	0.0983	0.0864	0.2126	0 202	0.1028

TABLE 17 COMPARISON OF VALUES OF Y

Size of Gear	$\frac{k}{6\,p}$	$\frac{\text{Table 1}}{\text{Lewis}}$	$0.124-\frac{0.684}{n}$
20T	0.0914	0.090	0.0898
30T	0.0995	0.102	0.1012
40T	0.1038	.....	0.1069
60T	0.1105	0.114	0.1126
80T	0.1109	.....	0.1154
100T	0.1128	0.118	0.1172
150T	0.1180	0.120	0.1194

$$W' = \frac{sft^2}{6\,l} \quad [5]$$

Also, by moments about center of gear

$$W'(r + l') = Wr \quad [6]$$

$$W' = W\frac{r}{r+l'} \quad [7]$$

Substituting value of W' from [7] in [5]

$$W\frac{r}{r+l'} = \frac{sft^2}{6l} \quad [8]$$

$$W = \frac{sf}{6}\,\frac{t^2}{l}\,\frac{r+l'}{r} \quad [9]$$

Say

$$W = \frac{sf}{6}\cdot k \quad [10]$$

where

$$k = \frac{t^2}{l}\,\frac{r+l'}{r} \quad [11]$$

By Lewis formula

$$W = spfy$$

Hence

$$spfy = \frac{sf}{6}\cdot k$$

$$y = \frac{k}{6p} \quad [12]$$

43 To find the factors y corresponding to the gear teeth of these tests, computations were made using the actual dimensions as given in Table 16. The results are formulated in Table 17, together with the values of y given in Table 1 of Lewis's original paper, and as computed from his formula

$$y = 0.124 - \frac{0.684}{n}$$

for 15 deg. involute, given in the American Machinist, June 22, 1893.

44 No single expression of the form $x - \frac{y}{n}$ can satisfy all of these values of $\frac{k}{6p}$. They call for a range from $0.124 - \frac{0.652}{n}$ to $0.124 - \frac{1.120}{n}$, but the expression $0.124 - \frac{0.684}{n}$, gives values for the factor of strength which do not vary by as much as 5 per cent in any case from the values determined from the actual tooth dimensions.

APPENDIX No. 3

DETERMINATION OF ARC OF ACTION

45 The tooth comes into contact, neglecting for the present those cases of gears with such a small number of teeth that there would be interference, when the addendum circle of the driven gear cuts the line of action. This position of the tooth is shown at c', Fig. 30. Contact between this tooth and its mate will continue until the addendum circle of the driver cuts the line of action, again neglecting those cases involving interference. This position is shown at c. The length of the arc of action $d'd$ is important as it governs the number of pairs of teeth simultaneously in contact.

46 The position of the tooth for pitch-point contact is shown at a. The corresponding point of tangency of the line of action and the base circle is b. It is obvious that b'' bears this same relationship to d; and b''' to d'. Hence, when the driven gear tooth travels from position $c'd'$ to cd, which is its complete period of action, the base circle will have rotated through the arc $b'''b''$. From the nature of the involute curve.

$$ab\ (= db'') = cb + \text{arc}\ bb''$$

Also

$$ab = ac + cb$$

Hence

$$ac = \text{arc}\ bb'' \qquad [13]$$

Similarly it can be shown that

$$ac' = \text{arc}\ bb''' \qquad [14]$$

Adding [13] and [14]

$$ac + ac' = \text{arc}\ b'' + \text{arc}\ b'''$$

Hence $cc' = arc\ b''b''' =$ length of line of actual action.

47 The cases involving interference may also be considered by reference to Fig. 30. If either gear has so few teeth that ab' becomes less than ac', the value of $b''b'''$ will, of course, equal $b'c$ (and not $c'c$). If both gears are low-numbered so that ab' is less than ac', and ab less than ac, the value of $b''b'''$ with equal $b'b$.

48 It is evident from Fig. 30 that the radius of base circle = the radius of pitch circle × cos 14½ deg. for the 14½ deg. involute system. Hence

Radius of pitch circle = radius of base circle × sec 14½ deg.

and

Arc of pitch circle corresponding to $b''b'''$ of base circle = $b''b'''$ sec 14½ deg. = length of arc of action.

49 The actual length of the line of action in any case may be readily computed.

50 Referring to Fig. 31, Oac is a triangle in which side Oa = pitch radius of driver r

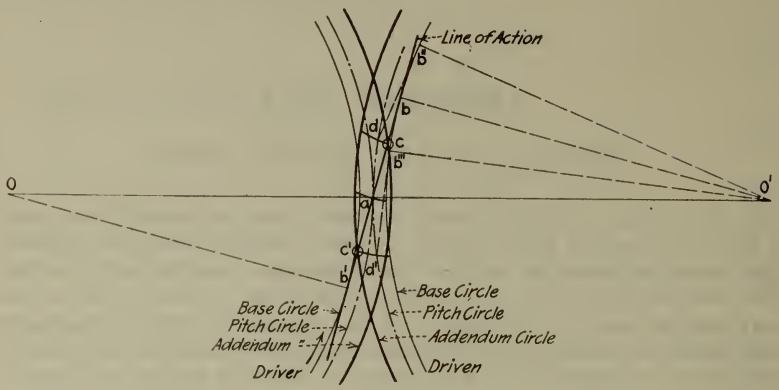


FIG. 30 SHOWING POINT OF INITIAL CONTACT OF TEETH

side Oc = pitch radius of driver plus addendum, $r + a$
angle Oac = 90 deg. + 14 deg. 30 min. = 104 deg. 30 min.

$$\sin x = \sin 104 \text{ deg. } 30 \text{ min.} \frac{r}{r+a}$$

hence angle x becomes known. Angle y = 180 deg. — (104 deg. 30 min. + angle x)

$$ac = r \frac{\sin y}{\sin x}$$

Similarly ac' may be computed for the driven gear.

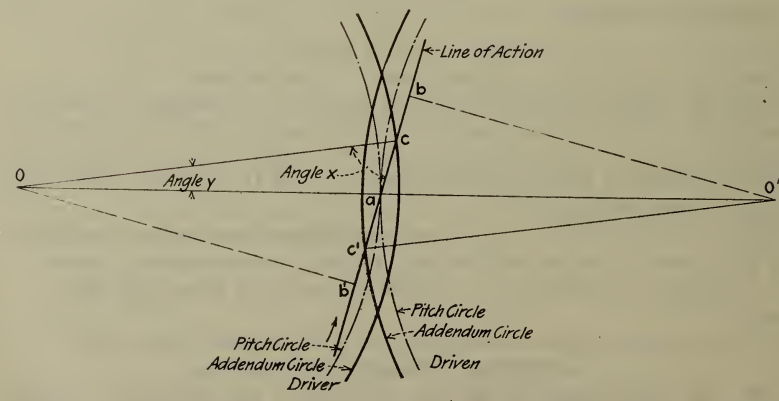


FIG. 31 DIAGRAM TO ILLUSTRATE LENGTH OF LINE OF ACTION OF TEETH

51 In cases involving interference it must be borne in mind that $ab' = Oa \sin 14\frac{1}{2}$ deg. and $ab = O'a \sin 14\frac{1}{2}$.

TABLE 18-a COMBINATIONS OF 10-PITCH GEARS

Gear, No. Teeth	*ab*, Inches	*ac*, Inches
20	0.25038	0.27182
30	0.37557	0.29605
40	0.50076	0.31208
60	0.75114	0.33246
80	1.00152	0.34495
100	1.25190	0.35345
150	1.87775	0.36637

The 20-tooth gear is the only one in the set used in these experiments where *ab* is used less than *ac*, i. e. where theoretical interference is involved.

TABLE 18-b COMBINATIONS OF 10-PITCH GEARS

Gear Teeth		Line of Action, Inches	Arc of Action, Inches
Driver	Driven		
20	30	0.52220	0.53938
20	100	0.52220	0.53938
20	150	0.52220	0.53938
30	30	0.59210	0.61158
30	40	0.60813	0.62814
30	60	0.62851	0.64919
30	80	0.64100	0.66209
30	100	0.64950	0.67087
30	150	0.66332	0.68514
40	40	0.62416	0.64469
40	60	0.64454	0.66575
40	80	0.65703	0.67866
60	80	0.67741	0.69970
80	100	0.69840	0.72138

In all these cases except those involving 20-tooth gears the line of action equals the sum of the respective *ac*'s. In cases involving the 20-tooth gears it equals the *ab* of the 20-tooth plus the *ac* of the 20-tooth.

52 From the values of *ab*, *ab'*, *ac* and *ac'* so computed the length of the actual line of action in any given case becomes known. This gives the length of the corresponding arc $b''\ b'''$ and multiplying this by sec $14\frac{1}{2}$ deg., the actual length of the arc of action is obtained.

53 Tables 18-a and 18-b give the results of such computations for the combinations of 10-pitch gears used in these tests.

APPENDIX No. 4

DERIVATION OF FORMULA

$$W = spf\left(0.154 - \frac{1.26}{n}\right)$$

54 Assuming the Lewis form of expression

$$W = spf\left(x - \frac{y}{n}\right)$$

as the general formula in which the values of x and y are to be determined. it is known that

$s = 39000$ in all cases
$p = 0.31416$ in all cases
$f = 1.0625$ in all cases
$W = 1184$ when $n = 20$
$= 1451$ when $n = 30$
$= 1608$ when $n = 40$
$spf = 13018$

When $n = 20$

$$1184 = 13018\left(x - \frac{y}{20}\right) \quad \text{[15]}$$

$$0.091 = x - \frac{y}{20} \quad \text{[15a]}$$

$$1.82 = 20\,x - y \quad \text{[15b]}$$

Similarly when $n = 30$

$$1451 = 13018\left(x - \frac{y}{30}\right) \quad \text{[16]}$$

$$0.112 = x - \frac{y}{30} \quad \text{[16a]}$$

$$3.36 = 30\,x - y \quad \text{[16b]}$$

And, when $n = 40$

$$1608 = 13018\left(x - \frac{y}{30}\right) \quad \text{[17]}$$

$$0.123 = x - \frac{y}{40} \quad \text{[17a]}$$

Subtracting [15b] from [16b]

$$30\,x - y = 3.36$$
$$20\,x - y = 1.82$$
$$10\,x = 1.54$$
$$x = 0.154$$

Substituting in [15a]

$$0.091 = 0.154 - \frac{y}{20}$$

$$\frac{y}{20} = 0.063$$

$$y = 1.26$$

Therefore

$$x - \frac{y}{n} = 0.154 - \frac{1.26}{n}$$

Check

$$0.154 - \frac{1.26}{20} = 0.091 \quad \text{when } n = 20$$

$$0.154 - \frac{1.26}{30} = 0.112 \quad \text{when } n = 30$$

$$0.154 - \frac{1.26}{40} = 0.1225 \quad \text{when } n = 40$$

DISCUSSION

Luther D. Burlingame. An apparent reason for the difference in results obtained by Professor Marx and what is shown by the Lewis formula is that of the distribution of the load: is it considered to be divided so as to be taken by several teeth, or is it figured to come all on one tooth? In a series of tests where all the conditions are ideal the load may be considered as distributed over several teeth. In actual use, however, it is often the case that one tooth must take the entire load, due either to material getting between the teeth of gears or to wear, faulty alignments, etc. Such conditions often bring the load not only on one tooth, but on one corner of a single tooth so that it seems proper that as far as the strength of arms and rims is concerned that this should be taken into consideration. The fact that in actual practice we often see teeth of standard gears of the sizes in question broken out where the arms and rims remain intact would go to prove this statement, and it is believed that this should have consideration in proportioning the arms and rims for general use. Whenever greater strength is required it is possible to use a coarser pitch which provides also a heavier rim and arms.

The tests recorded in this paper, carried as they are to the extreme, are interesting as showing possibilities and limitations under a uniform distribution of load. There are, however, many varying conditions and commercial gears are designed to meet the average of these conditions. At any rate it is safe as far as the strength of teeth is concerned to provide for one tooth taking the entire load.

R. E. Flanders. The work of Professor Marx appears to be the most important in the line of gearing investigation that has been made public since Mr. Lewis's paper was presented many years ago. The thing has been done that most needed to be done: that is, actual breaking tests have been made of gears running at heavy loads and high speeds. These investigations must have been expensive in time, power, and material, and this expense is probably what has hindered the carrying on of extensive experiments of this kind up to this time.

It has been generally recognized, I think, by designers that the Lewis formula gives an excess of strength when dealing with well made and nicely cut gears. In machine tool work it has

been customary to make the gears considerably smaller than Mr. Lewis's rules call for. As the author points out, Mr. Lewis left that matter open, admitting that there were no data for determining the effect of speed on strength. This paper gives reliable data of that sort for the first time.

I would like to see Professor Marx carry his experiments a little further so that the questions of the strength of gear teeth under running conditions may be settled as nearly as such a thing can be settled. With that end in view I would make the following suggestions:

The work of which this paper is a record naturally sums itself up in usable form in the formula in Par. 34. Here the safe working load is made a function of the modulus of rupture of the material used, the pitch of the gear, the width of face, the factor of safety, the number of teeth, a velocity coefficient, and a coefficient reckoned from the arc of action. There are certain considerations, some of them hinted at in the paper, which would make it seem wise to experiment further before writing out the formula definitely. The following factors may be mentioned as being still uncertain: dynamic strength of the material used; the fineness of pitch in cast-iron gears; ratio of the pitch to the width of the face; accuracy of tooth shape; and the advisability of using the arc of action in the calculation.

In regard to the dynamic qualities of the material, is it safe to use Table 6 for all materials? Does not the strength of a gear running at high speed depend more on the dynamic qualities of the metal than on the static strength? Would the coefficients of Table 6 derived from cast iron be correct when used for mild steel or when used for special heat treated alloy steels, such as used in automobile practice?

Is it safe to use comparatively fine pitch cast-iron gears as a basis for a general rule? The teeth of a fine pitch gear, close to the skin of the casting, might have quite different characteristics from the material in the interior, which is the material tested by specimens taken as were those shown in Fig. 6. It would seem that experiments should be carried out as well with more homogeneous metals.

Par. 36 indicates that further experiments should be carried out relating to the effect of the width of face. Is it wise to give the formula in its present incomplete condition, which makes strength directly dependent on the width of face?

It is also important to know how much the accuracy of the cutting effects the strength of the gears at high speed. All grades of accuracy are used in commercial work. To investigate this matter it might be possible to try two or three sets of gears, one made with the cutter set central, the next with the cutter set off center 0.002 in., and the other with the cutter set off center 0.003 in. or 0.004 in. The chances are that an investigation of this kind would show that a high premium is put on accuracy of cutting from the standpoint of strength. If this is so, it should be definitely known, though it is not practicable to include this factor in a formula.

In regard to the matter of the arc of action coefficient, taken from Table 14, I am strongly averse to using this. It is safest to follow Mr. Lewis's scheme of considering the whole load as acting on one pair of teeth. It is all very well to distribute the load between two or more teeth if properly cut and accurately mounted and *new* gears; but gears properly cut and accurately mounted, after being run under severe conditions for long periods of time, do certainly wear away from their true shape and tend to take the bearing on one pair of teeth at a time, this bearing being usually located near the pitch line where the least wear comes. It is therefore not safe in my opinion to make any provision for distributing the load over two or more pairs of teeth. This is an ideal rather than a practical condition, and is obtainable only with mechanism that is brand new.

I have indicated lines of experimentation that seem to be necessary before an authoritative formula can be given for the strength of gear teeth; but the amount of experimenting necessary to determine these various undetermined factors is well nigh staggering, both from the standpoint of time and of expense. Such experiments would have to extend over a long period of time. It might be possible to reduce the expense in the following manner:

Suppose that instead of generating a tremendous horsepower and then absorbing it all in a brake, the apparatus is so constructed that all the motor does is to furnish the power absorbed by the loss in friction. This is what Mr. Lewis did with the apparatus described in his paper.[1] As built the scheme has the difficulty that the load is fixed before the gears are started up:

[1] Trans. Am. Soc. M. E., vol. 32, p. 823.

they cannot be loaded while in operation. But it is conceivable that the mechanism might be designed so that the load could be applied gradually up to the breaking point of the gears. One way would be to take Professor Kenerson's transmission dynamometer and run it backwards; that is, use the fluid in the dynamometer for applying the load, instead of using it to measure the load. The conditions would be the same as in dynamo and motor testing, where the dynamo and motor run together, and no power is absorbed except that due to the losses. This would solve the difficulty of the heavy power expense, and the large size motor and brake required for any experiments with coarse pitch gears.

Furthermore, in the matter of experiments, why is it not a legitimate field for financial assistance from the Society itself? Gears are used in almost every conceivable branch of mechanical engineering. The strength of gears is one of the most important and one of the least understood calculations in the design of mechanism. If a fund is ever available for research work, here is a suitable field for it.

It is to be hoped that Professor Marx will be able to undertake further investigation along the lines suggested so as to add something to the splendid work he has already done in this matter of the strength of gear teeth.

W. S. Huson. My remarks may not bear directly on the paper, but they are based on analyses from experience coupled with formulae and are given to try to point out some hidden causes that formulae cannot cover. Assuming a gear must be made of a certain metal, if the teeth are not strong enough a coarser pitch is necessary unless some extraneous condition affects the strength.

Often more attention is paid to the selection of a cutter and the cutting of the blank than to getting shafts or bearings in proper alignment, so that the teeth do not bear evenly and are subjected to improper strain and wear, and in cases of close working fit, teeth bearing crosswise on their opposite sides create noise and undue wear. This is also the reason why wide face gear teeth sometimes break out and narrow faces do not, although the teeth wear away more in the latter case.

Gear cutters are often blamed for improperly mounted gearing, and while it is an absolute rule to have gear blanks of proper size

and to keep cutters sharp, the whole purpose is defeated if the same care is not observed in the final assembly of the intermeshed gears.

Involute teeth of 29 deg. in some cases under observation showed peculiar radial marks and a sliding action between the teeth. It has been suspected that there is a tendency created by the sliding to cause the teeth to pull away from their rim, and in the case of cast iron finally break.

Four-degree involute teeth do not seem to have these features. Apparently they roll together better and while they may not wear so long, they have a reasonable length of life. They are not so strong at the root of the tooth, but it is doubtful if teeth of given pitch break under a load, the 29-deg. teeth would be sufficient, hence, the safer method of the coarser pitch.

Teeth which stood up under belt drive now break when motor driven, due to the shock of quicker starting and in some cases too sudden stoppage by magnetic brake.

W. H. Diefendorf asked in view of the fact that the experiments were conducted with 10-pitch teeth at light load, with the load on the teeth doubtless much below the elastic limit of the material, if it would not be well to make experiments for the purpose of comparison where the loads were extremely heavy and to approach the elastic limit of the material.

J. Sellers Bancroft. When Mr. Lewis made his experiments on the duration of teeth at the shops of the William Sellers Company the lubrication of the bearing across the face of the gear was quite as important as the actual breaking strength of the teeth. It was the breaking away of the teeth in service that led him to adopt the formula.

William Kent said that Mr. Lewis's formula seemed to err, if it erred at all, in the direction of safety, which was a very good fault. His experiments and researches were made 20 or more years ago when the average practice in the foundry was not as good as it is at the present time, and the average cast iron, with which he had to deal, was probably not as strong as it now is. Table 7 gives the modulus of rupture in these tests as ranging from 37 to 47, showing that cast iron has considerable variability, which must be taken into consideration in all work on gears or anything into which cast iron enters.

C. H. Norton. We have been carrying on experiments to eradicate defects similar to what Mr. Huson has experienced in the printing press business. We have customers for grinding machines who desire to produce perfect cylinders, as determined by rubbing the ground work in a very perfect hole; and after a slight abrasion of the surface occurs, no error should be discernible. For years this seemed to be impracticable as it was found that any intermittent motion in the transmission of power to the machine, either through gearing, belting or other means, would cause the lines spoken of by Mr. Huson.

We therefore devised an instrument to detect the error of spacing in gears based on the principle used in detecting errors in surface plates, by rubbing one plate upon another. The plan consisted in having two axes of exactly the same center distance as those upon which the gears were to run. These axes were made to revolve at a uniform ratio as nearly as skill could make them. The gears to be tested were mounted on these axes and the surfaces of the teeth brought into rubbing contact, placed in a bath of oil with a minute amount of abrasive of the nature of Hindoostan stone, not emery, and run together for a short time with the axes rotated at the uniform ratio of speed by an independent means.

The connection between the axes was by worm-gearing, made as perfect as possible, not by hobbing the gears, but simply gashing them. The two worms were on the same axis and the two wormwheels in the same plane, having one of the worms adjustable on its shaft to enable the spaces of the teeth of the gears to be tested, to be brought into contact as mentioned.

It will be found in such a test that some of the teeth bear in one place and do not in another. I believe that these errors, while not of much consequence, and not practicable to overcome in any machine, can be remedied by running the gears together for a short period of time until all the teeth touch and by using the gears in the same relative position in which they were placed in the testing apparatus, unless the number of teeth should be odd, in which case more perfect gearing would be produced.

William C. Marshall. I have always understood in connection with the Lewis formula that the tooth was designed, not on the basis of continuous running, but so that when one tooth was brought against another tooth with the full force of a blow, the tooth could stand up. Professor Marx said nothing about

that for the teeth were running at constant speed. Another point to be considered before making a direct comparison of formulae is, are the lengths and depths of the teeth the same in both experiments?

Professor Marx says nothing about Mr. Barth's modification of the formula which does not give exactly the same results as Mr. Lewis. Mr. Lewis gives the pressure allowed on the tooth as a certain amount up to 100 ft. per min. for cast iron. Mr. Barth modifies that and gives an expression for the stress S on the tooth by means of the following formula

$$S\frac{600}{600+V}=S$$

where S = static stress and V = velocity in ft. per min.

Another point not brought out in the paper is whether or not the shafts were made exactly parallel. All the teeth broke off in a straight line along the bottom and none were fractured from the top down to the bottom on the other side, which is the case usually met with in cast-iron gears.

T. M. Phetteplace. Professor Marshall has brought out a point I have been considering, e.g., whether the teeth broke off as a whole tooth or not. The method of fracture would indicate that the shafts were parallel, but in most of the cases I have seen in actual practice the tooth broke in the corners, the shafts being out of line. If gears are to be designed to carry loads, I think they should be calculated to resist the load applied at the corner rather than along the entire width.

The pitch is so small that it applies to gears where it is usually difficult to obtain any idea of what the load might be, and that in such cases the trial method is the cheapest and best in securing gears of sufficient strength. Similar tests on gears of a No. 5 pitch, and under, would be of more value. I would suggest also that if a larger motor could be used in this apparatus that tests on gears with larger teeth would be more interesting, and possibly some tests on shafts that were not in line.

The question of tilting is always important in the strength of gears, but since that occurs so much in small gears it detracts from the value of the test. Another thing of importance in estimating the strength of gears from the viewpoint of a designer who wishes to figure on gears is that on small gears there is hardly any necessity for conducting such a test, since they are

cheap enough to test directly on the machines. The formula is of value only when it contains some symbol for the tensile strength of the material. This is an empirical formula for cast iron, although it contains something that is based on a physical test of cast iron.

I would suggest that the metal (Brown & Sharpe cast iron) of which the gears were made be tested, if possible. Perhaps a sample coupon could be poured or some test specimens made at the same time the gear was being made, so that the tensile strength of this cast iron could be known and in this way an expression be obtained that might possibly apply to other materials.

E. P. Lesley. From Par. 36 and Table 15 it appears that 10-pitch cast-iron gears of 17/32-in. face are not half so strong as the same pitch gears of 1 1/16-in. face. The writer observed several of the tests, the results of which are embodied in the paper, and was present when Professor Marx presented an abstract of the paper to the student society of mechanical engineers at Stanford University. When the above peculiarity was mentioned in the discussion, the writer advanced the following as a reason:

The cut-down gears were narrowed in face by turning on both sides. The outer metal of the casting, which is the stronger portion because of more rapid cooling, was thus wholly removed.

In order to test the reliability of the writer's contention, the following experiments were made: Several pieces of iron were cast in the university foundry 1⅛ in. square by 6 in. long. These were all cast in one flask and from the same ladle of metal.

A part of these castings were milled to ½ in. by 1 in., 1/16 in. being taken from two opposite sides and 5/16 in. from each of the other two sides. Others were milled to ½ in. square, 5/16 in. being taken from all sides of the casting. Still others were milled to ½ in. square, 1/16 in. being taken from one side of the casting, 9/16 in. from the opposite side, and 5/16 in. from each of the other sides.

These pieces were then broken in flexure with the following results:

Specimen	Average Modulus of Rupture, Lb.
½ in. by 1 in.	42,700
½ in. sq., center of casting	33,000
½ in. sq., side of casting	44,500

It would appear from these experiments that had the gears been cut to half width from one side only, they would have been half as strong as the original gears.

Wilfred Lewis. The paper under discussion is an important contribution to a subject which has been awaiting the light of experimental research for many years, and the author is to be congratulated as the first to demonstrate in a practical way the effect of speed on the strength of well-made gears. It is not surprising that the differences observed should be so little between the limits imposed, but these are hardly wide enough to be conclusive and it is to be hoped that further experiments will be made on heavier gear teeth, both cast and cut, at speeds extending to 3000 or 4000 ft. per min. Enough has been done, however, to demonstrate the possibility of producing gears of small pitch so well formed and spaced as to distribute the working load over more than one pair of teeth in action, but the question still remains whether such distribution can be safely relied upon in actual practice with gears of larger size and heavier pitches when not cut to the high standard of perfection attained by the Brown & Sharpe Manufacturing Company. It is well known also that small test specimens generally develop higher unit stresses than large ones and for this reason the values obtained for 10-pitch gears may be misleading when applied to heavier pitches. Another consideration is the allowance to be made for wear and also the effect of repeated and reversing loads upon the strength of the teeth.

It would, however, be rather difficult and expensive to repeat the experiments as made by Professor Marx on a much larger scale, and one important item in the work would be the power required to determine the effect of high speed on strength. The cost of power for such a test could, however, be reduced to one or two per cent of that used by the author if the load on the teeth were self-contained and not transmitted to a friction brake, the reference being to the apparatus used by the writer in measuring the friction of gear teeth as shown and described in his report to the Society in 1910.[1] The principle there employed could be used to advantage in a modified form of apparatus, whereby the load on the teeth could be adjusted as desired and broken gears could be readily removed and replaced. This is offered, however,

[1] Trans. Am. Soc. M. E., vol. 32, p. 823.

as a suggestion for further experiments rather than as a criticism of the good work already done.

It is true, as Professor Marx points out, that the working stresses suggested by the writer[1] as appropriate for different speeds, were based upon the factors of safety taken by Mr. Cooper from E. R. Walker of Newcastle-under-Lyme, but the working stress of 8000 lb. per sq. in. was adopted advisedly, and not, as suggested by the author, through an oversight of the well-known difference between the tensile strength of cast iron and its modulus for cross-breaking. My object in presenting certain assumed values for the permissible working stresses at various speeds was to use the best light available and keep on the safe side, and, as Professor Marx has very properly inferred, the values given were tentative only while awaiting the results of just such experiments as he has made. Probably Mr. Walker meant his factors of safety to apply to the breaking load, and from his point of view, perhaps, a factor of 3 would correspond to a fiber stress of 12,000 lb. per sq. in. in the accepted formula for cross-breaking, but from my point of view, as I will now explain, a factor of 3 should not allow a fiber stress greater than one-third of the ultimate tensile stress, or 8000 lb. for iron having 24,000 lb. tensile strength. The stress-strain diagrams for cast iron indicate pretty clearly that within safe working limits the stresses and strains are fairly proportional to each other, whereas when the ultimate strength is approached, the stresses do not follow the strains and the material begins to show fatigue. For this reason the accepted formula for fiber stress in bending applies fairly well to safe working loads but is very wide of the mark when ultimate loads are concerned. In other words, it is irrational from my standpoint to assume that the stress developed in bending can ever exceed the ultimate tensile strength of the material. The higher modulus for bending over direct tension is clearly due to a misapplication of the accepted formula which takes account neither of the modified relations of stress and strain near the breaking point, nor of the well-known shifting of the neutral axis in the cross-section. For convenience, however, the same formula is applied to all cases and no harm can result if it is borne in mind that the modulus for cross-breaking is simply a fiction designed to avoid the awkward necessity for a more ac-

[1] Proc. of Engrs. Club, Phila., 1893, vol. 10; also American Machinist, May 4, 1893.

curate expression of the truth. It varies with the cross-section of the material used and bears different proportions to the tensile strength for the same sections in various materials. For example, rectangular sections in cast iron may have a modulus for cross-breaking about 65 per cent greater than the tensile strength, while for round sections the modulus may be 100 per cent greater, and for flanged beams only 25 per cent greater. Conversely, if 3 is a proper factor of safety for direct tension, 4.95 would be right for rectangular sections, 6 for round sections, and 3.75 for flanged beams, because each of these factors when applied respectively to the section to which it refers leads to the same tensile stress in the material. It is, therefore, the safe working stress, rather than the breaking load, that determines the available strength of gear teeth, and for the reasons here given I would prefer to avoid any so-called factors of safety as misleading and dangerous.

At the same time it is evident that the breaking load must bear some definite relation to the safe load and that the former is more easily determined by experiment than the latter. What that relation should be is to be judged from the experimental researches made upon materials subjected to repeated and reversing stresses. A great many data on the subject are available, although not within my reach at present, but for gearing in general I would say that the maximum working stress at slow speed should not exceed the maximum reversing stress that may be indefinitely applied. Many gears, to be sure, transmit power only in one direction, while others act intermittently in both directions; but to be on the safe side I would not in any case exceed the maximum stress indicated for reversing loads, and from this I would deduct a reasonable margin for wear.

I am quite willing to admit that 8000 lb. per sq. in. may be too low for the working stress in small gear teeth when very accurately made, but there is a vast difference between the results obtained by the author and those quoted by him as made by the Brown & Sharpe Manufacturing Company. Their gear of 110 teeth having the same pitch and face showed a breaking strength of only 1060 lb. at a speed of about 72 ft. per minute, whereas the author's gear of only 30 teeth showed a breaking strength of 1850 lb. at a speed of 100 ft. per minute. The working strength of the B. & S. gear of 110 teeth would be 325 lb. by my formula, against 353.5 lb. as given by the author on the basis of a factor of safety of 3. My position is, therefore, pretty well

sustained by the B. & S. experiments made in 1892, but I would expect the gear specified to show a breaking strength of $325 \times 4.95 = 1609$ lb., instead of 1060 lb., and this value almost splits the difference between the values obtained by B. & S. and the author.

I am not at all surprised that the author has found a marked difference in strength for speeds of less than 100 ft. per minute. Originally I questioned the advisability of considering any speed as less than 100 ft. per minute because it would seldom occur in machinery driven by power, but afterwards at the suggestion of my friend, Carl G. Barth, my tabulated factors for speed were expressed by the formula $m = \frac{600}{600 + v}$, where $m =$ factor for speed and $v =$ velocity in ft. per minute at the pitch line of the gears. Mr. Barth adopted this expression for the influence of speed in his slide rule for the strength of gear teeth and subsequently he advocated an increase in my working stress from 8000 to 10,000 lb. per sq. in. as more nearly in accord with his observation and experience. Now it appears from the author's experiments that a new expression is needed for the value of m with a further advance in the safe working stress, but such changes should be made, if at all, with great caution.

The increase in strength for speeds in excess of 300 ft. per minute is an anomaly for which no adequate explanation has been given and it is premature to assume that such exceptional results point to any definite conclusion. The effect of centrifugal force is a negligible quantity in any practical case and the idea of an infinite speed causing a continuous load suggests the query *what load*, whether nothing at all between contacts or double the actual pressure as a live load; for no matter what the speed may be, any gear tooth, when running, is necessarily without load much the greater part of the time. So far from the speed being a somewhat favorable factor beyond 300 ft. per minute, cases might be cited where mortise gears of wood and iron have shown a working strength decidedly superior to that of two steel gears at the same speed, whereas at moderate speeds the steel gears would have shown much greater strength. For instance, in the discussion of my paper on Interchangeable Gearing, read before the Engineers' Club of Philadelphia in 1900, Mr. Graves mentioned the case of two cast-steel bevel gears with cut teeth running at 3900 ft. per minute under a fiber stress of 2100 lb. per sq. in., as estimated

for the whole load distributed along the end of a single tooth. The resistance of the material to abrasion under the load carried was satisfactory, but at different times teeth were broken out of the driving pinion which had 36 teeth, 5 in. pitch, 20 in. face. In the same room were three other sets of mortise gears with steel pinions having 33 teeth, 5½ in. pitch, 20 in. face. The teeth of these pinions were reduced to 2¼ in. in thickness at the pitch line and the stress per square inch upon them under the working load distributed along the end of a tooth was 2200 lb. per sq. in. The mortise wheels wore out in six weeks to two months and were regularly replaced, but their steel pinions never broke and at that time they had been in service about seven years, against five years for the all-steel gears which had broken repeatedly.

The static load in the first case for a working stress of 20,000 lb. per sq. in. would be 114,000 lb., while at 3900 ft. per minute the teeth broke repeatedly under a load of only 11,000 lb. Commenting on this case, Mr. Christie observed the greater difficulty of obtaining satisfactory results with bevel gears than with plain spurs, which of course must be admitted, but the reduction in strength at the higher speed is too great to admit of complete explanation in this way, and I think further confirmation of the surprising effect of high speed on the strength of heavy gearing can readily be adduced, although I cannot at this writing lay my hand upon the data I have in mind. I have no doubt at all that the strength of gear teeth is appreciably affected by the size of the mating gear, but I would not on this account consider the number of teeth in the mating gear as a factor to be reckoned with for interchangeable gearing where the smallest pinion in the system may engage with a gear of any size. The small difference in strength due to the length of the arc of action depends upon the system adopted in forming the teeth, and this difference would naturally be greater for teeth cut to the B. & S. standard than for involute teeth of greater obliquity in which the arc of action is more nearly constant for all combinations of gears.

To be on the safe side, I would prefer to assume that any gear might run with a 12-toothed pinion, and fix its working strength accordingly. I must also admit that the radial component of the driving pressure on gear teeth has some effect upon their strength. I referred to this point in my original paper which the author has done me the honor to quote so freely, but I did not

then and do not now attach much importance to it. In material like cast iron which is stronger in compression, the thrust between centers would add to the strength of the teeth; whereas in mild steel, which is stronger in tension, the thrust between centers would reduce the strength of the teeth. Some margin for wear is needed and a small variable of this class might reasonably be included in that margin by the adoption of a somewhat higher or lower working stress than would otherwise appear necessary or desirable. It is hard to reconcile the conclusions reached by Professor Marx with the data obtained from other sources, and although good ground has been shown for doubting the effect of speed to be so great as had been assumed, further corroboration, especially at high speeds and with gears of known irregularities in forming and spacing, is much to be desired. The impact due to such irregularities as ordinarily occur in cut or cast gears may be sufficient to account for the differences previously assumed in the strength of gears at various speeds, and until more data on this point are available it would seem unwise to adopt a new expression for the speed factor.

The experiments desired do not necessitate very elaborate and expensive apparatus, but they should be planned on a scale to include heavier pitches and much higher speeds. Professor Marx has certainly thrown a great deal of light upon the strength of gear teeth and I sincerely hope he will continue his experiments until more definite conclusions can be firmly established.

THE AUTHOR. Several participants in the discussion seem under a misapprehension in regard to the paper by Mr. Lewis to which the author has gone back as a foundation for his work and from which he has quoted so freely. The misapprehension evidences itself in those who speak of Mr. Lewis's formula having been based upon his experiments. Mr. Lewis's paper to which reference is made was not experimental. In a very neat and original manner, but entirely by theoretical analysis, he showed the necessity for considering the variations of strength of teeth of the same pitch due to their variation of form, and developed a mathematical expression for taking this into consideration. Up to the time of his publication, this important variation had been ignored by designers, and it may be said without exaggeration that his paper revolutionized practice in computing gear teeth in this country.

There being no adequate data available, he was compelled to choose values for strength *as affected by speed* according to his best judgment and in doing so he called attention to the lack of experimentally derived data. The aim of the author has been merely to begin the work of supplying the lacking data. So far as he knows this is the first systematic experimentation carried out to get such data for cast-iron gears, and no one can be more fully aware of the work that remains to be done. The necessary amount of experimentation is vast, as is indicated by the various suggestions made in the course of the discussion, and the entire field can be covered only by blocking out a relatively small portion at a time and exploring that thoroughly. Such a small field was selected in these tests.

It is to be noted, however, that the experiments recorded here had a range of pitch speed from zero to 1000 ft. per minute. Although the results could not be plotted in Figs. 4 and 5 (because the teeth did not break) it is significant that, in experiment No. 02, a 20-T, 10 pitch, C.I. gear, running at 1005.3 ft. per minute, bore, without fracturing, a load of 800 lb. at the pitch line. Also (experiment No. 14), at a speed of 731.36 ft., the 20-T pinion carried, without breaking, the load of 1024 lb. at pitch line. Such results, while not definitive, indicate clearly that the strength of these 20-T pinions at 1000 ft. per minute was more than 50 per cent of their static strength, and at 731 ft. more than 60 per cent. Just how much more we can not tell, as, unfortunately, these loads stalled the largest motor available. In this same connection, it is significant that (experiments Nos. 48 and 50) the 17/32-in. face gears, 30-T meshing with 30-T, showed the same strength at 508 and 580 ft. pitch speed; and that 1 1/16-in. face gears (experiments Nos. 49, 51 and 52), 30-T meshing with 30-T, broke with a load of 1408 lb. at a speed of 508 ft., and could not be broken (stalling the motor) with a load of 1323 lb., at 580 ft. In the same way light is thrown on the continuation of the curve of Fig. 5 by experiments Nos. 30 and 31, in which 30-T gears meshing with 40-T safely carried loads of 1632 and 1600 lb. at speeds of 480 and 501 ft. respectively, clearly demonstrating that the curve must lie above these points, and to that extent corroborating the high results of experiments Nos. 32, 33, and 34. The apparent rise of the curves of strength after a speed of 300 ft. per minute was passed is something that should distinctly have fur-

ther investigation before it is accepted. But the demonstration seems adequate that in a range of pitch speeds from 0 to 1000 ft. per minute the falling off of allowable pitch line load need not be more than 50 per cent, provided the conditions otherwise, as to reversals of stress and suddenness of application of load, remain the same.

It is true, as Mr. Lewis and others remark, that these tests were carried out on quite small gears and that small test specimens in general show higher unit strength than large ones; but the much more significant fact must not be overlooked, namely, that *the general behavior of large and small test specimens of similar proportions is the same.* In other words, small specimens serve to show the *laws* of variation as well as large ones. If such laws are determined with small specimens first, relatively few tests are necessary with the larger specimens to check the results and to obtain their proper allowable unit stresses. It is the author's conviction that large gears under similar tests will show the same general behavior, under variations of speed and arc of action, as are shown here by these small gears, although the probability of irregular results due to weak spots will be much increased and the modulus of rupture probably found to be lower the larger the teeth tested.

The question has been raised of the degree of parallelism of the shafts. The first apparatus which the author designed for these tests, in 1902, provided means which would insure practically perfect parallelism, but for reasons of difficulty and expense of construction this was abandoned and the simple apparatus described in this paper used in its stead. The degree of accuracy employed in the construction of the apparatus and the care used in setting up the tests were no greater than those which prevail in any good machine shop, and it was thought that the results would have greater practical value for this very reason.

The author is glad that Mr. Burlingame raised the question of the load coming on one corner of a single tooth. The same point is made by others in the discussion. On this matter Mr. Lewis said in his original paper:

> In estimating the strength of teeth, the first question to be considered is the point or line at which the load may be applied to produce the greatest bending stress. In the rules referred to, the load is sometimes assumed to be applied at the pitch line, sometimes at the end of the tooth, and sometimes at one corner,

but in good modern machinery, the agricultural type excepted, there is seldom any occasion to assume that the load is not fairly distributed across the teeth. Of course, it may be concentrated at one corner, as the result of careless alignment or defective design, in which the shafts are too light or improperly supported, and, for a rough class of work, allowance should certainly be made for this contingency, but, in all cases where a reasonable amount of care is exercised in fitting, a full bearing across the teeth will soon be attained in service. It must be admitted, however, that on account of the inevitable yielding of shafts and bearings, even of the stiffest construction, the distribution of pressure may not be uniform under variable loads, and that the assumption of uniform pressure across the teeth is not always realized in the best practice. To what extent it is realized I shall not attempt to estimate, but in general practice, where the width of the teeth is not more than two or three times the pitch, the departure can not be regarded as serious. The conclusion is therefore reached that in first class machinery, for which the present investigation is intended, the load can be more properly taken as well distributed across the tooth than as concentrated at one corner.

The author's judgment coincides with that of Mr. Lewis, and is corroborated by his experience both in shop practice and in these tests. To get further light, several garages have just been visited, the workmen interviewed, a considerable number of broken gears examined, and all the testimony was to the effect that, where teeth broke at all, i.e., were not battered nor crushed nor worn out, they broke across the entire width of face, "as square across as a pipe-stem," one of the garage proprietors expressed it. The question to ask ourselves is this: In the case of gears whose width of face is not more than three times the circular pitch, have we actually seen cut, cast-iron, spur gears with any of their teeth broken off cornerwise? Is not this rather a tradition, which has come down to us from the days of rough cast gears, and which it is time to discard when we consider modern cut gears and the prevalent degree of accuracy in shop practice?

Another fallacy is that of the teeth of one gear striking the teeth of another "with the full force of a blow." Modern gears are cut and adjusted to run without backlash. If there is no backlash, it is important to note that there can be no blow delivered to the teeth of one gear by those of its mate. It is true that the load comes on and goes off in a single instant, i.e., a "suddenly applied," repetitive stress, but it can not be a load "applied with velocity." No formula of which the author has knowledge attempts to provide for anything more than these suddenly applied, repetitive stresses at varying rates of pitch

speed and continuous running. For such cases as true intermittent gears, i.e., with omitted teeth, a blow is possible; and to compute the resulting stress in such cases it is necessary to know the energy possessed by the impinging mass, the equivalent mass attached to the driven gear, its acceleration at the instant of impingement, and the elastic yielding of the teeth and attached parts.

The next point generally discussed is that of the advisability of taking the influence of the arc of action into consideration. There can be no question of this influence. The curves of Fig. 9 show that the fact is beyond dispute. That Mr. Burlingame, Mr. Flanders and Mr. Lewis all advise ignoring this factor, is worthy of serious consideration. Nevertheless, the author feels constrained, for the reasons indicated in Par. 26, to believe that this factor acts just the same in practice as it did in these experiments and that it should no more be ignored than variations in strength due to changes of tooth profile. Mr. Flanders states that the bearing on worn teeth is usually located near the pitch line. This would then become the point of application of the load (as in Fig. 8) and not the end of the tooth (as in Fig. 7), and it is exactly this condition of affairs which the factor a provides for. The factor is present to show that, if the arc of action exceeds the pitch arc, a single tooth can not get caught in its weakest position to carry the whole load. Without this factor we provide not merely that a single tooth carry the entire load but also that it carry it in its weakest position, as illustrated at A of Table 8, an assumption contrary to fact. If something uncrushable gets between a pair of engaging teeth, there being no backlash, a wreck is inevitable in any case. It could only be avoided if the teeth were strong enough to bear the stress arising from bringing the machine to a full stop instantaneously. In this connection, the importance of the tests suggested, involving the accuracy of spacing, deserves emphasis.

Mr. Flanders asks whether it would be safe to use Table 6 for all materials. That is a question which can be answered only by experimenting on the other materials. However, cast iron is one of the poorest metals we have for resisting anything akin to tensile shock and it would seem safe to assume that no other material would show a more rapid falling off in "dynamic strength" with increase in suddenness of application of load. The author doubts if it would be possible to carry on breaking

tests like these with more ductile materials such as steel or bronze. It would seem that the carrying capacity of such gears would be found to be limited, not by the cross-breaking strength of the teeth, but by their capacity to resist localized crushing and abrasion. This will probably be found, as in rollers, to vary with the square of the least radius of curvature of the tooth profile, pointing to the desirability of using, for these materials, a form of tooth having the flattest possible profile.

Professor Lesley's explanation probably gives the main reason why the half-width gears did not show half the strength of the full-width ones. The known superior strength of cast iron near its surface points further to the desirability, if it were not too hard on the cutters, of rough-casting the teeth for cutting, rather than cutting from the solid blank even for small pitches, where extra strength is desired. The question of best width of face needs further investigation. There are several conflicting factors beside those mentioned heretofore. The narrower the face of the original casting, the stronger the material is apt to be. On the other hand, considering the form of the cross-section of the tooth, the less the width of face in proportion to the pitch the lower the probable modulus of rupture for the same quality of material. It would seem that the best that can be done, in the present state of our knowledge, will be to continue to assume that the strength varies in direct proportion to the width of face.

Mr. Huson brings out an important point. It is true of gears, and many other machine parts, that rigidity may actually favor rupture in cases of shock. The capacity to resist shock depends not upon strength alone but upon the resilience of the part; and, in this physical characteristic, the elastic yielding is a vital factor. The relative strength, *at high speeds*, of stiff, unyielding teeth and thinner, more flexible teeth, is one of the fields of experimentation yet unexplored. It should be impartially investigated, for strong conflicting opinions are held in regard to it. It may be stated, however, without fear of contradiction, that the heavier the attached parts, the stiffer the shafts on which the gears are mounted, and the greater the rigidity of the transmission throughout, the greater will be the induced stresses in the teeth due to shock. Increasing the shock-resisting power, by increasing the elastic yielding, is an important and not sufficiently recognized principle in all design.

Professor Marshall asks whether the proportions of the teeth of the gears experimented with are such as to make them justly comparable with those used by Mr. Lewis in the development of his formula. This is fully answered in the affirmative in Par. 24 and Appendix No. 2 of the paper. The author said nothing about Mr. Barth's formula because this is based upon Mr. Lewis's assumptions and, therefore, it seemed preferable to go back for purposes of comparison to the original source. A number of such formulae have been proposed, but since they are all based upon merely arbitrary assumptions it did not seem necessary nor advisable to complicate matters by discussing them in a paper whose sole object was to supply a few facts where we had previously been limited to speculation.

Professor Phetteplace will find the author's excuse for making these experiments stated in the early paragraphs of this closure. Physical tests of the gear material, made on specimens cut from the gear themselves and having approximately the dimensions of the teeth at their section of fracture, will be found in Par. 22 and Table 7 of the paper. He does not explain why he holds the unique view that a flexure formula should contain a symbol for tensile strength rather than one for the modulus of rupture in flexure. One is also left in doubt as to just how the tensile strength tests on one material are to give an expression that might apply to other materials, and why the suggested formula is empirical, rather than rational with empirically derived values for its terms.

Mr. Lewis has made an excellent statement of the limitations of the theory of flexure. That the author is in substantial agreement with him can be seen by referring to the middle of Par. 29. It was just because of recognition of the variation of the value of the modulus of rupture with form of cross-section, that the test specimens were made of the same cross-section as the teeth. As Mr. Lewis aptly says, we make use of a fiction in order to avoid the necessity of using a very much more awkward expression of something nearer the truth. That is exactly what all theories and so-called natural laws are, convenient generalizations put in usable form for working purposes of partial and near-truths. It may be that these teeth, being subject to compound stresses, actually fail by shear, according to Guest's theory of stresses. It is more than likely that the apparent stress of 36,000 lb, per sq. in. corresponds to an actual tensile fiber

stress of but 24,000 lb. But the author cannot follow Mr. Lewis in his next step. If he wishes to limit the actual tensile stress to 8000 lb. this would permit the use in the formula for flexure of an apparent stress of 12,000 lb., with the same margin of safety. The author would not be understood as recommending an apparent stress of this high value for cast iron. What he considers to be guiding values are quoted from Mr. Christie, with the conditions of their applicability, in Par. 4. The corresponding "actual" tensile stresses would then range from 6000 lb. (for slow speed and steady load) down to 3000 lb. (for high speed and reversed stresses). In a broad way it is true, that, for reversed repetitive stresses, just one-half of the safe value for repetitive stresses between zero and an upper limit, is allowable.

The case which Mr. Lewis quotes,[1] in which the thinner steel pinion teeth, meshing with wooden mortise-wheel teeth, did not break, while the thicker steel pinion teeth in mesh with other steel teeth did break, appears to be a very good illustration of the value of greater flexibility where suddenly applied loads are concerned. This is confirmed by Mr. Graves' statement in discussing the quality of the steel of the various gears: "As is to be expected, the softer has resisted breaking the longer." That is to say, the more ductile material, of course, showed the greater shock-resisting power. The all-steel gears had 1/16-in. backlash to begin with, which permitted blows as an accompaniment of the greatly varying loads, and how much greater the backlash became, due to flattening of the tooth surfaces because of these blows, there is no record given. Teeth for such service should have the least possible curvature of profile. In the author's judgment, a somewhat flexible torsional-coupling between the driven gear and its shaft would have saved these breakages.

The case, moreover, is one of bevel gears, and not directly comparable to the spur gears under discussion. Mr. Graves pointed out that the effect of power transmission in these cases was to throw the axis lines of the gears out of position, moving them in opposite directions, and resulting in end contact of teeth and concentrated loads. He correctly stated that in this particular the question of maintaining bevel gears is decidedly more of a problem than that of spur gears. And that in the use of the

[1] Proc. Engineers Club of Phila., vol. 18.

latter small end motions of carrying shafts produce no effect; while the wearing of bearings, as it occurs slowly, adjusts itself. The just conclusion seems to be that these bevel gears failed because, on account of the allowable play, there resulted faulty alignment and severely localized stress. It does not seem demonstrated that the speed was a vital factor in their failure.

A better case for comparison would be the steel spur gears described by Mr. Christie in the same volume.[1] They had a pitch of 4.92 in., as opposed to that of 5½ in. of the bevel gears; and had a pitch speed of 2500 ft., as compared with the speed of the bevel gears of 3100 ft. at their mid-point. They had the severe duty of connecting the engines to a rolling mill, yet they bore, with perfect satisfaction, three times the load per inch width of face that fractured the bevel gears. They were of 16-deg. involute type and had no backlash.

In closing, the author wishes to express his appreciation of the interest shown by the contributors to the discussion, and his thanks for the generous words of commendation of his work that have been spoken. In particular he wishes to express his gratification that Mr. Lewis, in whose work he found his source of inspiration, has deemed the paper worthy of the serious consideration shown in his discussion. He recognizes the justice of the criticisms made on the limited area covered by the investigation and the value of the suggestions for further experimentation. He has added to these himself in this closure and has refrained from mentioning many more which naturally have occurred to him in his long period of interest in the subject. There is no danger of exhausting the field. Those who have carried on similar investigations in connection with their regular work know how slow the progress is and how little provision there is in this country for such work. Meager as these results may be, they represent holding to the idea for full ten years. The first apparatus was designed in 1902 and the paper completed in 1912. Of course, such slow progress would not be necessary if the Society adopted some such plan for aiding investigations as Mr. Flanders suggests, or if we had National Experiment Stations, adequately financed, something similar to our Agricultural Experiment Stations, along the lines of those in Germany. Is there not something here for the Society to consider seriously?

[1] Proc. Engineers Club of Phila., vol. 18, p. 59.

The author hopes to be able to supplement this work some day by means of larger and better apparatus devised along the lines suggested by Mr. Flanders and Mr. Lewis.

No. 1383

SYMPOSIUM ON CEMENT MANUFACTURE

DEPRECIATION FACTOR IN THE COST OF PRODUCING PORTLAND CEMENT

By Frederick H. Lewis, Birmingham, Ala.

Member of the Society

At the moment the question of the depreciation factor in the cost of producing portland cement is commercial rather than technical. Its urgent importance as a business proposition overshadows its scientific interest. The manufacturers of cement have so notably failed to define the cost of production, that few of them have a surplus or depreciation fund of any kind to deal with. In recent years numerous plants have yielded returns little if any greater than bare factory cost, and the commercial rating of the industry as a whole has suffered. The value which a technical discussion of depreciation may have, therefore, will consist chiefly in defining for the commercial end of the business the elements which should be considered to make up a satisfactory price for a barrel of cement.

2 The portland cement industry in America is but 37 years old. The first factory was built at Coplay, Pa., in 1875. This plant grew to considerable size and importance during the 25 years ending in 1900. But during the latter part of this period units of considerable size were one by one discarded as having reached the limit of their useful life. The early developments of the American, Atlas, Alpha and Lawrence companies were similarly discarded. It is the author's judgment, based on personal observation only, that the average useful life of plant units discarded in this way did not exceed 10 years, and that it must

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have required 15 to 20 cents per bbl. to cover the depreciation factor. Fortunately at that time prices carried an adequate margin of profit.

3 It is quite likely that the present plant is more nearly standardized and that it may suffer less from obsolescence. But this is not certain; and the simple fact is that we have no sufficient experience to justify this conclusion. The development of production has been so rapid during the seventeen years just past, that if we assume increased output to be directly represented by the new plant (which is not strictly true), then we shall find that the average age of the plant units which will produce this year's output is seven years. We have then no sufficient industrial history to establish our data. We are forced to exercise our best judgment.

4 Fifteen years ago an English manufacturer stated to the writer that the repair cost in the manufacture of cement would average six pence per bbl. (12 cents). Subsequent experience verified this to be a reliable estimate. Today the figure is probably nearer 15 cents because the mechanical equipment has been much elaborated in recent years. If it is assumed that the necessary capital required is $1.50 per bbl. of annual output, then the ordinary repairs in the manufacture of cement amount each year to 10 per cent of the entire investment. This is not depreciation; it has nothing directly to do with depreciation. It is ordinary repairs. Depreciation "is the necessary annual installment of the annuity which will amount to the first cost of the property at the expiration of its useful life." The cost of ordinary repairs is merely the first factor to be considered in estimating the useful life of the plant. The high cost of repairs indicates rapid depreciation, or a short useful life.

5 The second factor to be considered is obsolescence—that form of depreciation due to improvements in methods or in machines which makes it necessary to modify plant and scrap machinery which is still in good running order. As already indicated we have few data in regard to this. It may be accepted as definitely true, however, that few plants could be successfully operated today with the equipment and arrangement they had ten years ago. The very few which might make such a claim, can do so only because either by chance, or by reason of greater foresight, they happened to be designed along the lines which later

progress has followed. Plants which are now ten years old or more and which are still operated, have generally had large expenditures put into them for enlargements and for improvements in details; and in spite of this are operating at some disadvantage by reason of obsolete machinery. Kilns and power plants in particular have changed notably in design, and the economy of heavier machinery, and of labor-saving equipment has continually claimed recognition. Enlargements and modifications of plant have more often been forced upon manufacturers than desired by them. Enlargement beyond a certain point is no longer desirable, but has frequently been accepted as the only alternative to closing down the plant. Such forced enlargements represent, therefore, in considerable part, depreciation pure and simple, the gradual replacement of an obsolete plant which will be as gradually discarded. Yet the cost has in most cases been paid for out of new capital. The writer has not sufficient data to be able to express an opinion on the cost due to obsolescence during the period from 1900 to January 1, 1912. No one familiar with the facts will doubt that it has been large.

6 A third factor to be considered in estimating depreciation is loss of profits due to shutdowns for repairs and renewals. This is often a considerable sum, by reason of repair periods extending over three, four, or even six months. If there is no surplus fund to carry interest and overhead charges during such periods, then the business has not fully provided for the expenses due to depreciation.

7 There are then three principal factors to be considered in estimating a depreciation factor: (*a*) wear and tear; (*b*) obsolescence; and (*c*) the carrying charges during repairs.

8 The ordinary procedure of cement plants is to limit their cost data to a monthly cost sheet. This deals simply with factory cost and usually, and quite properly, with only a part of the factory cost. The real and only useful purpose of these monthly cost sheets is to enable operating officers to institute comparisons, locate leaks, and cut down regular operating cost items. Unfortunately, and quite generally, the cement people have made the serious commercial blunder of regarding these factory cost sheets as really representing the cost to be kept in view when making prices. An immense industry has been rendered unremunerative for five years past by this unfortunate blunder.

9 A factory cost sheet to be useful for comparison of cost items from month to month and from year to year, should exclude all occasional or extraordinary expenditures. If items of this character were introduced at random as they occurred, the value of the cost sheet would be destroyed. Such occasional or extraordinary expenditures, which are really depreciation charges, must be provided for annually by appropriating a sum for plant renewals to be accounted for separately from the cost sheet. This appropriation can be an arbitrary one, or may vary from time to time as the superintendent's report may indicate the renewal requirements.

10 It is to be remarked here that it is usual to deal with depreciation as an annuity payment at compound interest. The propriety of figuring the annuity at compound interest for cement plants is open to question. A cement plant is not at all like the one horse shay which went to pieces all at once, and nothing first. On the contrary there must be a considerable annual depreciation fund for immediate use, and which can have no interest accumulation. Not to gainsay an accepted method, depreciation should be provided for in two items: (*a*) an annual plant renewal charge, and (*b*) a depreciation charge. The accounts of the U. S. Steel Corporation follow this method. There is a regular depreciation fund which is 3 per cent of gross earnings, and a construction or plant renewal fund of an approximately equal amount, but which varies more or less as earnings vary.

11 Coming then to a method of figuring depreciation, let us make the following assumptions:

a Net capital investment = \$1.50 per bbl. of annual output, or 1.5 cents per bbl. = 1 per cent return on the capital.

b True depreciation to be based upon 25 years' useful life at 4 per cent compound interest. This equals 2.4 per cent per annum or 3.6 cents per bbl. of cement.

c Plant renewal to be based on 25 years' useful life, but without interest accumulation. This equals 4 per cent per annum, or 6 cents per bbl. of cement.

This basis assumes that each of the several items of the plant will be renewed once in 25 years, and that at the end of this period of 25 years the entire plant will have reached the end of its useful life. The author is prepared to agree that this

basis is low. But if the general argument is good, then the figures may be modified by exchange of views and data.

12 Assuming a factory cost of 60 cents per bbl., a remunerative selling price would then be estimated about as follows:

	Cents per Bbl.
(*a*) Factory cost including all ordinary repairs	60.0
(*b*) Depreciation fund	3.6
(*c*) Plant renewal fund	6.0
(*d*) Administration and selling expense, say	10.0
(*e*) Interest at 6 per cent on investment	9.0
Selling price	88.6

It will require about 90 cents per bbl. to come out even in this assumed case. It will require a higher average price if the business is to have a credit balance outside of depreciation.

13 On some such basis as this any manufacturer can estimate a safe selling price, using the actual facts of his own books for capitalization, factory cost, administration and selling expense, etc. If this is done most manufacturers will find that current prices have been too low in recent years, and still are so; many will find that an excessive capitalization must be scaled down if the business is to continue. And some will find themselves facing conditions which make it impossible to continue the industry.

DISCUSSION [1]

G. S. Brown.[1] A cement plant where both buildings and machinery are subjected to a very heavy wear, should be considered from a very different standpoint than the ordinary plant where repairs are light and buildings are likely to last almost indefinitely. In the latter case any items of repair must of necessity come at long periods, while in a going cement plant the item of repair is continually before us, and it seems useless to consider a definite percentage of the original cost as a measure of the depreciation. The bulk of the investment in a cement plant is in the machinery, and one authority says: "If repairs are made so as to preserve machinery in first-class condition so that the work performed is of the same quality as when machinery is new, it does not seem equitable to charge off a percentage for depreciation."

This exactly describes the condition which must obtain in a cement plant if its output and cost are to be kept satisfactorily. Certainly no cement plant can compete today if the machinery in it is not kept in topnotch condition. It is, I believe, the practice with most manufacturers to take care of this by charging regularly each month to the cost of manufacturing all expenditures for the upkeep of the machinery and buildings, and as long as this is done there should be no need of additional charge for depreciation.

In a plant with which I am well acquainted there are machines which have been in continuous service for the last 15 years. These machines today do more work than they did 15 years ago. As a matter of fact, however, there is probably not a single part of the original machine now in service; indeed many parts have been replaced several times annually. Nevertheless from the standpoint of an appraiser, the machine is quite as valuable as a new one. Instances which show how the necessity for keeping machinery up to its maximum precludes the need of a depreciation account, are by no means as rare as the average engineer may suppose; indeed there are few machines in a cement plant not entirely obsolete, of which a similar statement may not be made. We realize, of course, that in the

[1] In connection with the paper by Mr. Lewis, a paper by H. Struckmann on Depreciation and Obsolescence, presented at the Annual Meeting of the Society in December 1911 and published in abstract in Volume 33 of Transactions, was discussed at this meeting.

event of a mill discontinuing operations then the value of the plant has depreciated enormously. Few companies, however, would be willing to carry a depreciation account on their books sufficient to reduce the value at which a going plant was carried to a sum which would represent its value in the event of its no longer being operated.

When, however, the question of obsolescence enters we have a very different condition, particularly in a business where there is as yet no standard practice and no standard machinery. Machinery considered modern today is obsolete tomorrow and the only possible way that a mill can keep in competitive condition is ruthlessly to dismantle machinery which has become obsolete and replace it with more economical modern machines. It hardly seems possible that so large a percentage of plants as Mr. Struckmann cites can be operating in the present condition of the industry with out-of-date machinery.

The question of how the expenditures for renewing obsolete machines and buildings shall be handled is no doubt a very difficult one. The fact that charging such expenditures to cost of manufacture increases this item unreasonably and unjustly in particular months has led many corporations to put on their books accounts such as Improvement Account, Extension Account, Renewal Account, etc., with the result that ultimately the sum carried on the books as the value of the plant has been increased to an unwarranted degree. Some cement companies have a rule which requires that any expenditure for machinery or buildings which in any way replaces those heretofore in service, irrespective of whether the new machinery increases output or not, must be charged to repairs. When this rule is invariably followed the question of obsolescence must be satisfactorily cared for, although many experts will insist that an adjustment be made transferring a percentage of the expenditures to capital to equate for any increase of output or decrease of cost. The above method is open to the same serious criticism that it increases the cost abnormally at certain times and does not give a fair opportunity to compare them from month to month.

It has always seemed to me that the most reasonable method for handling the replacement of obsolete machinery is to have a certain specific charge made monthly to the manufacturing account against every barrel of cement manufactured, this charge to be more than sufficient to take care of, in the course

of a year, those expenditures which experience has proved must be made annually in order to replace machinery which has become out of date. Against the fund which would be derived from this course would be charged all expenditures for renewals, and in this way the result obtained would be exactly the same as if all such charges were originally made to the manufacturing account. The effect would be, however, to distribute these charges more uniformly throughout the year, and give a very much more satisfactory cost for the purpose of comparison. Any system of accounting, however, which considers any portion of the repairs, or the expenditures for the replacement of obsolete machinery and buildings as capital expenditure, will ultimately cause trouble for those who manage the concern.

R. C. Carpenter stated that he made a very careful study of many of the points in this paper. The percentages of selling price as figured out, are practically 50 per cent more than the manufacturing cost. He did not believe that many manufacturers outside of the cement industry could make money with a selling price not more than 50 per cent greater than the cost of production. That percentage seems to be a very low amount for overhead charges, administration, interest and depreciation. If what Mr. Lewis states is correct, the earnings in the cement industry do not look so unfavorable as compared with other industries.

As to obsolescence, the speaker pointed out that in the cement industry, grinding machinery, for instance, had a remarkably long life, both in view of the time that it has been kept in use, and of the improvements in that kind of machines. In regard to kilns there has been a very great change due to improvements in the last few years. On the whole he is more in accord with Mr. Brown in his discussion than with Mr. Lewis. The grinding mills have a long life, but to keep the machines in running order a large and constant repair account is necessary. His important question is whether this large repair account should properly be charged to depreciation, or to the operation of the plant. Professor Carpenter thought that the proper way to find the true depreciation was to ascertain whether the amount charged for that purpose made the capital account really different from its true valuation. If the plant becomes of less value year by year, and there is no money to cover this loss, then there is a depreciation which has not been taken account of.

There should be an increase in cash corresponding to the loss in value, if the depreciation is taken care of, and if the value of the plant is not kept up to the original cost. The depreciation may be accounted for, as Mr. Brown states, by an actual charge per barrel against manufacturing cost.

R. K. Meade raised the question of the relation between the sinking fund and depreciation account: if a plant is built with bonds, and charges off every year for sinking fund, should it maintain also a depreciation fund? As regards the question of obsolescence with particular regard to the cement industry, then one had to do more with the problem of turning out cement in very much larger quantities than with actual changes of process. In some cases, especially in the West, the question of the supply of raw material, particularly limestone, is also of importance.

P. H. Wilson claimed that Mr. Lewis was wrong in stating that the net capital invested is only $1.50 per bbl. In most cement mills in the United States it runs between $2 and $3 per bbl., and if his calculations are taken on the basis of $2.50, there will be added considerably more than 50 per cent to the net cost of depreciation and of selling a barrel of cement. As to interest, Mr. Lewis estimated it at 6 per cent on the investment. The figures should be pretty nearly doubled, with the exception of the administration or selling expenses, which are not affected by the capitalization of a mill to prove an attractive investment.

A few years ago Mr. Mallory made up some interesting figures on the cement industry in the United States, and found that this industry turned its capital over at a less rapid rate than any other industry in the United States. It takes the cement man about three years of actual work to turn over his capital and therefore, instead of the interest being charged at a flat rate, it should be increased to cover the fact that the capital is turned over so slowly.

Joseph Brobston explained that while the amount of stocks or bonds issued on a plant may be $2 or $3, the amount of money really invested per barrel is very conservatively estimated by Mr. Lewis at $1.50, the great trouble with the industry being that in many cases plants have $2 or $3 per barrel capitalization, due to the fact nothing has been charged off for depreciation, and all replacements of machinery have been added to the

capitalization. If a sinking fund were maintained, as suggested by Mr. Meade, that would take care of a part of the depreciation charge, and by the time it was necessary to replace the machinery, the bonded obligation would be reduced to such an extent that more money could be obtained without over-capitalization. Ten cents per barrel should cover both depreciation and obsolescence, but at that figure the machinery would have to be kept in good order all the time, with the repairs being charged into the monthly cost.

To arrive at a proper depreciation charge one should start in at the quarry. There are some plants where the cost of replacing their stone would be practically a negligible item, but a great many companies have a limited supply, and these companies should charge off the amount it would cost to replace it, and not what the land originally cost.

H. Struckmann. At a meeting of the Association of American Portland Cement Manufacturers, the writer brought up the question of depreciation charges, and suggested a minimum of 0.10 per barrel, which, according to Mr. Lewis' views would be equivalent to about 7 per cent of the capital invested in the plants. The fact that the current repair charges in a cement plant amounts to between 20 and 25 per cent of the total manufacturing cost, indicates clearly that the depreciation going on is considerably higher than in the average industrial plant, and that if anything, 7 per cent on the invested capital, or 10 cents a barrel is too low a charge to cover depreciation.

The following classifications of depreciation charges, which are adopted in the plants with which the writer is connected, may be of interest:

	Per Cent of Reproductive Value
Factory buildings	5
Factory machinery	12
Power plant equipment	10
Electric light system	7
Railway tracks	7
Quarry rails and equipment	15
Tools and fixtures	10
General construction, fences, retaining walls, etc	7
Water works and pipelines	10
Oil storage tanks	7
Dwellings	5

The above various percentages represent on the basis of capital invested per barrel output, as outlined by Mr. Lewis, approximately a flat rate of 15 cents per barrel, and the writer's experience has shown that it is none too heavy a charge if the investors in portland cement plants are to be protected.

Mr. Lewis' statement that carrying charges during shutdowns of the plant on acount of repairs are properly chargeable to the depreciation fund, in the writer's opinion, is not correct. Interest, overhead, as well as other carrying charges of similar nature, have nothing to do whatever with depreciation and should be taken care of not by the depreciation reserve fund, but by a contingent reserve fund, created not only to take care of the above items but also to cover charges accruing while the plant is idle on account of the market conditions, as well as extraordinary accidents like explosions, floods or other causes beyond control.

ELECTRIC DRIVING IN A CEMENT PLANT

By R. K. Meade, Baltimore, Md.

Member of the Society

14 At the plant of the Tidewater Portland Cement Company, Union Bridge, Md., all the machinery, including conveyors and elevators, is driven by individual motors. The central power plant consists of three 23 by 46 by 48 in. compound condensing Corliss engines made by the Wisconsin Engine Company, and direct connected to General Electric 800-kw. generators supplying 3-phase current at 550 volts and 25 cycles; a reserve 150-kw. unit is also at hand. A frequency meter is installed to control the cycles and by them the speed of the mills. A recording meter controls the efficiency of the power plant independently of that of the whole system and, in combination with automatic coal scales, gives all the information required.

15 Throughout the mill itself recording watt-meters are installed in the various departments. This allows a record to be kept of the power used in each department. Readings of these are taken twice a day, with the change of shifts, and any unusual rise in power per barrel in the department is looked into at once. This results in low power consumption because as soon as a rise occurs, it is instantly traced to its source. When the mill was first started, a great deal of trouble was experienced with the motor drives. Most of this was due to faulty arrangement. In some cases, the motors were much too small for the load on them, and in others, large motors were pulling very light loads. The result was that the large motors were running at a very low load factor and the small motors were giving a great deal of trouble from overheating and even burning out. In order to remedy the trouble a test was made of every motor on the plant when operating under normal conditions, and, as the result of this test, a redistribution of the motors was made, and in all about 40 motors were changed. Much of the trouble with the motor drives was thus eliminated. In changing from a line drive to a motor drive and in installing a motor drive in a new plant, much

thought should be given to the size of motors employed. When the Tidewater was built very little information was available relative to the horsepower required by conveyors and elevators operating in cement mills and most of the locating was done by guess rather than by science. Manufacturers generally are apt to under-estimate the power required by their machinery. At the same time, if too liberal allowances on their figures are made and motors much too large are employed, a low load factor will result, with the need of additional generating machinery in the power plant or the use of a few synchronous motors about the plant.

16 One objection to the employment of small individual motors for a group of elevators and conveyors is that the small motors are likely to stall from an overloaded conveyor or elevator, while with the larger motor the surplus power is sufficient to overcome this overload. This point against the individual drive, however, is to my mind, more than offset by the absence of belting and countershafts in the mill.

17 At the Tidewater plant, the conveyors and elevators are all driven by back geared motors of the squirrel cage type and the writer's experience with these extends to motors up to 30 h.p. The gears and pinions after a year's service show almost no wear. The gears are encased in a dustproof gear case, partly filled with a cheap grade of heavy oil. Power is transmitted from motor to conveyor or elevator by means of a sprocket and chain.

18 The Fuller mills at Tidewater are all driven by vertical motors. The 42-in. mills are driven on the raw side and in the coal mills by 75-h.p. motors and the 54-in. "Dreadnaughts," on the clinker side, by 200-h.p. motors. The 200-h.p. motors were the first vertical motors of this size ever built and at first gave some trouble because of improper lubrication, which has been remedied. Experience has shown that it is far superior to the old quarter twist belt formerly employed with mills of the Fuller-Lehigh and Griffin types.

19 At Tidewater the motors driving the Fuller mills are installed with indicating ammeters which are placed in full view of the mill attendant and when the indicator on the ammeters to the 200-h.p. motors points to 175 amperes, the attendant cuts off the feed to the mill and so avoids a choking of the latter.

20. A compressed air line runs from the compressor in the quarry to the mill, with branches to various departments. This is used for blowing out the rotors which are for this purpose lifted

out of the motors by means of an overhead trolley and chain block. The motors are protected by no-voltage and overload relays.

21 The following system of taking care of the motors was used at Tidewater:

22 The motors are in charge of men appointed for that purpose and known as "motor-men." It is their duty to start all motors after a shutdown for repairs or for a choke in the mills or elevators. This avoids burn-outs. They also make thermometer readings of every motor once a day and inspect the motors hourly. At the end of the day, they make a written report of the condition of each and every motor in the plant and advise what repairs, if any, are necessary. They also give the number of hours each motor has been in operation during the shift and report on all shutdowns, stating whether these were due to the motors or to the machines they were driving.

23 This enables an exact history of each motor to be kept and for this purpose each is given a number, starting with zero. All records are kept in a special ledger known as the "motor ledger," a page being devoted to each motor. At the head is placed the plant number and the figures obtained from the manufacturer's name plate as to type, rating and shop serial number. Any repairs are charged on this page and at any time an exact figure as to the cost of upkeep of any motor can be obtained.

24 Table 1 gives a list of the motors at the plant of the Tidewater Portland Cement Company, as arranged, September 15, 1912.

TABLE 1 MOTORS, TIDEWATER PORTLAND CEMENT COMPANY
AS NOW ARRANGED FOR 300 BBL. DAILY CAPACITY

MACHINE	MOTOR, H.P.
Air compressor	100
Incline chain hoist for cars	22
Car dump	12
Limestone crusher, No. 9, gates	150
Limestone crusher (2) No. 6, gates	50
Pan conveyor from the above, back geared	5
Elevator from above to dryer stock boxes	20
Belt conveyor from above to stone storage, back geared	5
Belt conveyor from limestone storage, back geared	5
Shale crusher, No. 6, gates	50
Pan conveyor from above, back geared	10
Elevator from above to stock boxes above dryer, back geared	15
Screw conveyor above dryer stock boxes, back geared	10
Belt conveyor to shale storage, back geared	5
Belt conveyor from shale storage, back geared	5

TABLE 1—*Continued*

Machine	Motor, H. P
Limestone dryer 7 ft. in diameter, and 50 ft. long	30
Shale dryer 7 ft. in diameter and 50 ft. long	15
McCaslin conveyor, limestone, back geared	10
McCaslin conveyor, shale, back geared	10
Ball mills, Krupp No. 8 (3)	50
Elevator from two limestone ball mills to mix bins, back geared	20
Elevator from shale ball mill to mix bins, back geared	10
Screw conveyor from mixing bins to scale, back geared	10
Screw conveyor from mixing scales to elevator, back geared	10
Elevator from above to the pulverizer stock boxes, back geared	10
Fuller mills, 42 in. (8) raw materials, vertical (8)	75
Fuller mills, 54 in. (2) raw materials, vertical (2)	200
Screw conveyor from mills, back geared	5
Rotary coolers, 5 ft. diameter and 40 ft. long, back geared	10
Screw conveyor over pulverized bins, back geared	10
Screw conveyor from kiln stack dust chambers, back geared	5
Fans, motors are direct connected, high speed	10
Screw conveyors feeding pulverized coal to burners, back geared	2
Kilns, 8 ft. in diameter and 125 ft. long, variable speed	30
Rotary coolers, 5 ft. diameter and 40 ft. long, back geared	10
Inclined pan conveyors from kilns, back geared	5
Belt conveyor from above, back geared	5
Elevator from above, back geared	10
Belt conveyor from clinker storage, back geared	7½
Belt conveyor under clinker storage, back geared	10
Bucket elevator from gypsum, back geared	5
Elevator from clinker and gypsum scales, back geared	20
Fuller mills, 54 in. clinker (10), vertical	200
Screw conveyor from below to bins in stock house, back geared	10
Belt conveyor to stock house, back geared	10
Screw conveyor from bins, back geared	30
Elevator from above to packing machine boxes	7½
Bates valve baggers (two on one motor)	30
Bag tying machine	2
Belt under coal trestle, back geared	30
Crushing rolls and elevator from above, back geared	5
Ruggles-Coles dryer, coal, back geared	7½
Elevator from dryers, back geared	7½
Meade dryer, coal	10
Fuller mills, 42 in. coal (3), vertical	75
Elevator and conveyor from above, back geared	10
Coal tunnel at power house conveyor, back geared	5
Elevator from above, back geared	7½
Drag conveyor from above, back geared	10
Stokers in boiler room (all on one motor), back geared	10
Machine shop (entire equipment)	15
Bag cleaning wheel	15
Motor driven exciter	150
Hydrated lime plant (entire equipment)	100

THE PREVENTION OF MISSED FIRES IN BLASTING

By W. H. Mason, Stewartsville, N. J.

Member of the Society

25 A frequent cause of serious accidents in cement manufacture is the explosives used in the quarries. Many of these are due to what is commonly called "missed fires," that is, bore holes which have been loaded with explosives and which fail to explode when detonated. Formerly when the holes were exploded by time fuses this was probably more frequent than at present, when it is common practice to explode a series of holes by an electric battery, fuses being used only to explode the small "plug holes" or dobies, for the purpose of breaking large stone which has already been blasted from the ledge.

26 A "missed fire" is always a source of danger. There is no known way of unloading it safely and when an attempt is made to drill another hole near it, so as to blast it out, the drill bit not infrequently comes in contact with the dynamite in the loaded hole with disastrous results to the driller. When the driller is fortunate enough to get his hole in without accident, it is loaded and blasted. This usually throws the powder from the "missed fire" hole among the debris and frequently when the latter is being loaded, a laborer will strike it with some tool and cause an explosion. The only way to avoid this danger is to prevent the missed fires. Newspaper clippings collected by the Du Pont de Nemours Powder Company show that there were 59 people killed and 157 injured in the United States in the last three years, due to missed fires.

27 For about eight years the Edison Portland Cement Company have been exploding all churn drill blast holes with a 500-volt direct-current power circuit. Many of these blasts are made of from 12 to 20 holes, each loaded with 500 to 800 lb.

of 40 and 50 per cent gelatine powder, and during this time there has been only one missed fire hole. It was impossible to find the cause of this failure, but it was attributed to a poor connection from the exploder leads to the bus wires. Before this time, they had been using the common push-down batteries and it was a usual thing to have several missed fire holes per month.

28 One of the quarries of the company is so located that no power circuit is available. For a year or two the blasts were set off by push-down batteries. After having had several missed fire holes, and some resulting accidents, they installed a small 250-volt direct-current generator, driven by a steam turbine. This has been in operation several years and there have been no missed fires since its installation.

29 Gelatine powder is the most difficult of all the dynamites to detonate and it is a well-known fact that if exploded by a powerful detonator, the full effect of the powder is obtained, while if improperly detonated, much of the effect is lost. For this reason and also on account of the danger and expense which always accompany a missed fire hole, the writer believes it is well to use too many exploders, or electric fuses, of high power than not enough. In a 6-in. bore hole, running 60 ft. deep, not less than three exploders should be used. One of these should be directly at the bottom, so that if when loading the hole with dynamite, a stone should slide in from the side and block it off, it would still be possible to explode the dynamite below, and thereby avoid the danger from the otherwise unexploded dynamite; the other two are distributed one near the center and the other near the top. All are connected directly across the 500-volt or 250-volt direct-current line, so that all exploders are in parallel and receive the full voltage of the power circuit. By this method any number of holes may be exploded at one time.

30 The cost of the extra exploders is always a small proportion of the cost of the dynamite used and the writer believes that this expense is warranted by the increased detonation power of the exploder, as well as by a sort of insurance against missed fires, which are always expensive aside from their potential danger.

31 Fig. 1 shows the method of connecting the wires of the electric fuses to the bus wires. The argument has been advanced frequently that the insulation on the leads of the electric fuses

is not sufficient to withstand the high voltage. This, no doubt, is true, but experience shows the fuse in the exploder blows

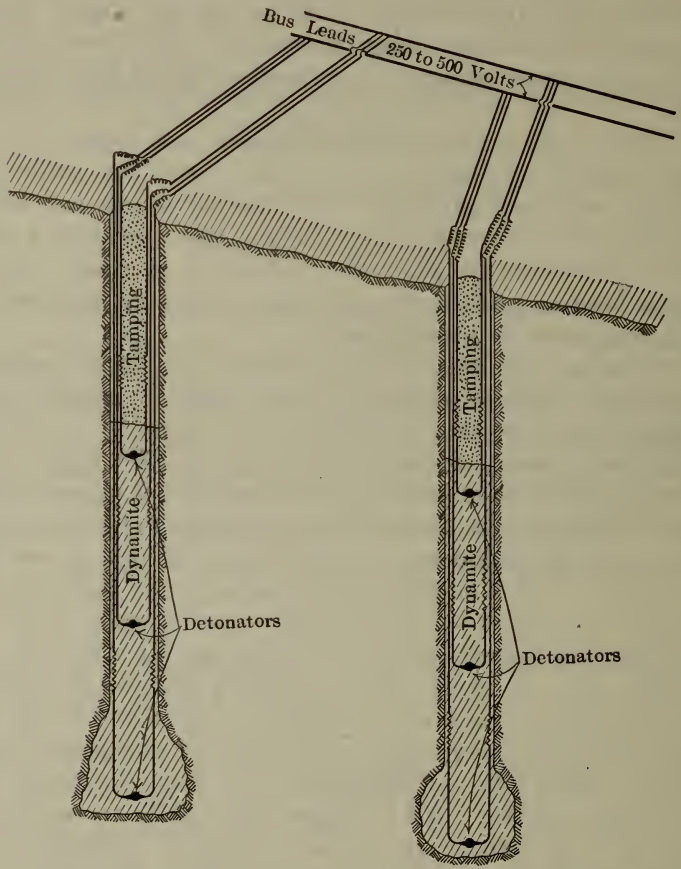

Fig. 1 Exploder Leads and Connections to Bus Wires

while the insulation is breaking down, and it is then a matter of no importance whether the insulation fails or not.

THE DETERIORATION AND SPONTANEOUS COMBUSTION OF GAS COAL

By Perry Barker, Boston, Mass.

Junior Member of the Society

32 In the storage of coal for use in cement manufacture the risk due to spontaneous combustion must receive serious consideration. Spontaneous combustion is caused by slow oxidation, principally of carbon compounds in an air supply, which is sufficient to support the process of oxidation but not sufficient to remove entirely the heat thus produced. The majority of cement manufacturers use gas coal slack for burning, having an average analysis as follows:

Moisture	3.00
Volatile matter	33.50
Fixed Carbon	55.50
Ash	8.00
	100.00
Sulphur	1.50

33 While the amount or composition of the volatile constituents of coal, or the percentage of sulphur, gives no direct indication of the liability to spontaneous combustion, fuels of this type have been known in a number of cases to heat in storage to such an extent that ignition and serious damage due to fires have resulted. The total loss of heat value due to storage without spontaneous combustion is comparatively small. However, the results of deterioration which has proceeded to the ignition point are more serious, involving the change in the character of the coal due to loss in volatile matter, abnormal percentage of moisture contained in the fuel after quenching as a last resort, costs of moving the coal in an endeavor to arrest the spread of the fire after the coal has become heated in certain localities in the piles and financial loss from fires directly traceable to this cause.

34 A number of erroneous reports have been submitted as to the extent of loss in heating value due to the deterioration of coal exposed in storage. These reports contain various figures on the loss in heating value, ranging from 10 to 50 per cent. Some results of tests on German gas coals have been submitted which state that these fuels have sustained an average loss of 1.7 per cent of heating value per week.

35 The diverse nature of the data which have been published on this matter, together with the importance of the subject, have prompted extensive experiments by the U. S. Bureau of Mines, the Illinois State Engineering Experiment Station, and several private interests to determine the exact nature of the reactions involved and to study methods for prevention. The processes of deterioration from which spontaneous combustion ultimately results may be best stated as follows:

a The escape of combustible gases when fresh surfaces of coal are exposed after removal from mine workings. Experiments on coals having the analysis described indicate that the loss of combustible gases in this manner is approximately equivalent to the volume of the coal. The heating value of these gases amounts to 0.1 to 0.2 per cent of the calorific value of the coal. The detrimental feature of this action does not lie in the small loss in fuel value but in the dangers incurred by the escape of inflammable gases in mines and the oxidation which accompanies the evolution of these gases.

b The absorption of oxygen from the air with consequent evolution of heat. This reaction is conceded to be the primary factor in the process of coal deterioration and production of heat in piles and is caused by the direct union of the oxygen of the atmosphere with unsaturated carbon compounds in the coal matter. Freshly mined coal is very susceptible to this reaction and absorbs oxygen with great avidity at the time of evolution of combustible gases.

c The size of coal, particularly of the type used for cement manufacture, is a primary factor in promoting deterioration. Fine coal brings larger surface exposure in contact with the atmosphere and accelerates the process of oxidation in the entire mass.

d The presence of moisture in abnormal percentages, particularly if coal in storage is wet and dried alternately, accelerates the processes of absorption and oxidation. The extensive experiments made by the Illinois State Engineering Experiment Station have shown that the activity of these reactions is increased by wetting the samples under test.

e External sources of heat, even though moderate in degree, tend to promote the reactions which produce spontaneous combustion. Heat produced by external sources may originate from contact with steam pipes, boiler walls or floors. This heat may also be traceable to the climatic conditions.

f The importance of the oxidation of sulphur compounds in the process of deterioration is a question of some contention among the authorities on this subject. Although the total heat evolved from this source is not large in comparison with that produced by the several sources noted above, it will undoubtedly promote to some extent the other reactions.

36 A number of experiments have been conducted upon high volatile American coals to determine the actual loss in heating value during storage, together with observations on the slacking or reduction in size produced by the deterioration. Universal opinion is now to the effect that such coals will lose from 1 to 3 per cent of their heating value in one year. A large percentage of this loss results within from three to six months after the coal is placed in storage. It is apparent that the loss in heating value is not the most serious consideration in the question of coal storage. However, the ultimate results of deterioration which has proceeded to the ignition point, although not determinable in financial units, demand that certain precautions be taken to prevent the procedure of this action to a point where ignition will result. The methods of prevention can best be divided into the following general headings:

a Elimination of combustible gases before storage.

b Storage in piles not over 10 or 12 ft. deep with maximum points the same distance from exterior of storage.

c The removal of fine coal so far as possible, storing only the screened lump. If a mixture of lump and fine coal is stored the various sizes should be evenly dis-

tributed to prevent the formation of air passages through segregated portions of large sizes.

d Ventilation by pipes should not be attempted and the admission of air by infiltration through brick work or porous foundations should be prevented.

e Exposure to weather conditions, where the coal is subject to heavy rains and subsequent drying from the sun or other sources of external heating such as steam pipes or heated brick work, should be avoided.

f The elimination of heating by the prevention of action through submerged storage. The effect of a large percentage of moisture introduced by submerged storage has been materially lessened in a plant recently constructed in which coal is stored but partially submerged. It has been found that the exposed portion of the piles maintains a sufficiently high percentage of moisture through capillary attraction to prevent serious oxidation.

g Prevention of dangerous dust and gas explosions as well as spontaneous heating caused by the storage of powdered coal. Coal for burning clinker should be pulverized at the furnace doors and only at the rate required for immediate use.

36 While the extent of the loss in heating value through deterioration of high volatile American coals, such as are used in cement manufacture, may not be of serious importance, the heating effects of the reactions incident to this deterioration demand attention to the designing of coal storage and to the necessary procedure in arresting deterioration which has begun through lack of proper precautions.

DISCUSSION

E. Y. Sayer referred to a recent bulletin issued by the United States Government on the deterioration of gas coal and stated that the facts set forth in this bulletin indicated that the depreciation taking place in gas coal was very rapid at first, commencing immediately after the coal was first mined, but the rapidity of deterioration was greatly reduced as time went on, and after a year or so it became almost a fixed quantity.

If this is the case he thought that a great many companies were making a mistake by periodically or at frequent intervals using up their storage. It would seem that storage should be looked upon as merely a matter of insurance and that after a sufficient amount of coal had been placed in storage it should not be used until absolutely necessary to draw on it. Every time that coal is taken out of storage, we are simply pocketing the loss due to the early rapid depreciation; whereas, if the coal is allowed to remain in storage, this loss will be incurred only at long intervals of time.

The Author in answering Mr. Sayer's questions, stated that, if a pile starts to fire, the first alternative would undoubtedly be turning over, provided the combustion had not proceeded to such an extent that it was impossible to get close enough to the source of heat. He had found a number of pile fires where the coal had become incandescent, and there the only alternative was flooding. The method of using closely covered storages in order to prevent spontaneous combustion the speaker found to be a very dangerous one. A fairly large volume of CO_2 would be produced with the possibility that the process of heating would proceed to such an extent that large volumes of combustible gases would also be liberated, leading to explosions. He cited a case where fire was caused by closely covered storage. A pile of ten or twelve thousand tons of coal, of exactly the character now under consideration was placed in outside storage. Snow fell to the depth of say two or three inches and instead of leaving the coal which was already covered with snow open, the whole storage was covered closely with rough lumber, laid tight to the pile, and down to the ground, with the result that there was no ventilation whatever. The coal was wet, and serious fires started.

Another of Mr. Sayer's suggestions was that, from a commercial standpoint, storage should be looked upon as a matter of

insurance. The stored coal should not be used up until it is absolutely necessary to draw on it, because the greatest deterioration takes place in the first few days after the coal is mined, and changing the coal in the storage often means simply pocketing the loss due to the yearly depreciation. Mr. Barker pointed out, however, that it is very dangerous to leave coal in storage for a great length of time. The two problems of deterioration and spontaneous combustion, while interdependent, are somewhat different. The loss in heat values may proceed rapidly in freshly mined coal, and ultimately reach a minimum, but an action may have been started to such an extent that it will proceed violently with resulting ignition. Besides, as shown in the paper, there is depreciation in the actual gas making value of coal when placed in storage.

The question of moving coal which has reached a certain temperature is of great importance. A temperature of 140 to 150 deg. fahr. is considered sufficient cause for taking steps to move the coal, preferably digging a trench of such width and length that the source of the high temperature can readily be exposed without covering it with any fresh coal.

During the presentation of his paper, Mr. Barker referred to the subject of fires and described a method of determining the temperature at the interior of a pile by means of a ¾-in. auger containing a maximum thermometer. The auger is screwed to sections of iron rod of slightly smaller diameter and can be driven in the pile to a distance of about 20 ft. and it requires about 20 minutes to record a maximum temperature. The result with lump coal is not very successful, but with the Eastern Pennsylvania bituminous coals that come to the market it afforded a ready means for driving the thermometer to the source of the trouble.

In reply to a question, Mr. Barker stated that the possibility of the heat produced by frictional resistance in driving the auger into the coal affecting the amount registered by the thermometer had been fully considered. It seemed, however, to be a factor of little consequence, because the auger is driven at the rate of 20 ft. in two or three minutes. Mr. Sayer suggested that this question could be settled fully by inserting an auger in a pile of coal of which the temperature is precisely known; then the recording thermometer in the auger would show just what happens.

No. 1384

NECROLOGY

GEORGE R. BABBITT

George Rodney Babbitt, president and treasurer of the American Oil Company of Providence, R. I., died at his home in Edgewood, R. I., on June 29, 1912. Mr. Babbitt, who was born at Berkley, Bristol Co., Mass., on June 27, 1842, received his education in the schools of that locality and in Fall River, Mass. As a young man he followed the sea for several years, making a voyage to California in 1860, then a tedious undertaking.

In 1862 Mr. Babbitt came to Providence and entered the employ of the Burnside Rifle Works, becoming a sub-contractor on parts of the work. When the works closed down three years later to be refitted for building locomotives, he spent a period in the Pennsylvania oil fields, and the following year became associated with William A. Harris, the Harris-Corliss steam engine builder. With this concern he filled the positions of machinist, foreman and superintendent, and invented and patented several improvements, among them a piston packing, for which he received a medal at the Cincinnati Industrial Exposition in 1874. a releasing gear and an extended piston.

In October 1888, he was tendered the position of superintendent of the heavy ordnance department of the Government at Washington, D. C., which, however, he declined. He bought an interest in the American Oil Company of Providence in 1892, with which he was associated at the time of his death. He was frequently called upon to act as a consulting and mechanical engineer for large manufacturing concerns, both for new work and for the best methods of repairing. In 1894 he patented an automatic barrel filler.

One notable achievement of Mr. Babbitt's was his plan for floating the steamship Paris which had grounded on some rocks while crossing the Atlantic, and which he proposed to the owners after reading an illustrated newspaper description of the accident.

RALPH P. BADEAU

Ralph P. Badeau was born in Brooklyn, N. Y., October 26, 1886, and received his early education at Rutgers College Preparatory School of New Brunswick, N. J., and at the Battin High School of Elizabeth. He was graduated from Stevens Institute of Technology in the class of 1909, and immediately afterward entered the employment of Purdy and Henderson, structural engineers of New York. While with them he had charge of the design of the McNeill Building, Chicago, Ill., and of the crane runway for Inland Steel Company. In November 1910 he became assistant superintendent of the A. & F. Brown Company of Elizabeth, N. J. where he was employed until his death, July 20, 1912.

ERNEST S. BOWEN

Ernest S. Bowen was born at Levanna, N. Y., May 28, 1858, and died at Geneva, N. Y., April 27, 1912. At an early age he went to work for the J. A. Spencer Iron Works, Union Springs, N. Y., where he received his first training and practical experience in mechanics. Realizing the advantage of a technical education, he entered Cornell University and worked his way through, graduating in the class of 1890. Immediately after graduation he entered the employ of the McIntosh Seymour Engine Company, manufacturers of high-speed engines, Auburn, N. Y., of which he soon became assistant superintendent. In 1895 he embarked in business for himself, forming a partnership with Walter L. Fay, also of Auburn, for the manufacture of bicycle parts. After five years they sold out, but re-formed a partnership for the manufacture of marine engines under the name of Fay & Bowen. As this business grew they added to it the manufacture of motor boats, which necessitated their moving to a location with a water front. They were attracted to Geneva and in 1904 the business was incorporated under its present name of the Fay & Bowen Engine Company and it has steadily grown until it is one of the leading concerns of its kind in the country.

W. F. BUCK

W. F. Buck, superintendent of motive power of the Atchison, Topeka and Santa Fé Railway, with headquarters at Chicago, Ill., died while traveling through San Bernardino County, California, on January 31, 1912.

Mr. Buck, who was born at St. Andrews, N. B., on June 9, 1863, received his education in the public schools of that community, entering railway service in 1882 as an apprentice in the shops of the Intercolonial Railway, where he received a practical mechanical training. In 1886 he severed his connection with this road and entered the employ of the Northern Pacific Railway as roundhouse foreman at Staples, Minn., and in 1895 he was promoted to the position of general foreman first at Missoula, Mont., and later at Helena, Mont. Mr. Buck steadily advanced in his profession, becoming master mechanic in 1902 of the Rocky Mountain Division of the Northern Pacific Railway, with headquarters at Missoula, and in 1904 of the Arizona division of the Atchison, Topeka & Santa Fé Railway, at Needles, Cal. Several years later he was appointed mechanical superintendent of the Eastern Grand Division of the same company, with headquarters at Topeka, Kansas, and in 1908 entered upon the work in which he was engaged at the time of his death.

Mr. Buck was always deeply interested in his profession and was the designer of a number of practical mechanical devices for railroad work, including a locomotive superheated and a locomotive external throttle.

FREDERICK H. CRABTREE

Frederick H. Crabtree was born at Ebbw Vale, England, October 5, 1863, and died in Anaconda, Montana, August 3, 1912. His early education was obtained in the Cardiff High School, and in 1876 he entered as an apprentice the Blaina Iron Works of Blaina, England, where he remained until 1881. Subsequently he was employed by the Thames Iron & Shipbuilding Company, London; Crossley's Woolen Factory, Halifax; Blaina Blast Furnace Works; Ebbw Vale Steel Works; and as chief engineer in charge of the Pyle Works coal washing and coking plant, Blaina.

Mr. Crabtree came to America in 1888 and accepted a position as master mechanic with the Rock Springs Coal Company of the Union Pacific Railroad. In 1890 he moved to Anaconda, Mont., where he held a number of responsible positions, among which was the supervision of the installation of the machinery in a 50-ton gold mill for the Republic Reduction Company, Republic, Wash. The last position he filled was as assistant superintend-

ent of power plants of the Washoe Smelter Company, from which he resigned three years ago.

Mr. Crabtree patented a number of devices. In 1887 he was granted an English patent on improvements on a rotary steam engine; in 1904 on a pneumatic massage apparatus; in 1905 on an automatic cut-off steam valve; in 1910 on a paper fastener and on an auxiliary rim and tire protector.

ALBERT W. DANFORTH

Albert W. Danforth who died at his home in Lowell, Mass., on April 25, 1912, was educated in the Lowell public schools and began his professional career as mechanical engineer for the Lawrence Corporation of that city. A few years later he became mill and constructing engineer for the Hamilton Corporation, and subsequently designer and superintendent of construction on machinery for the United States Cartridge Company, and mill and mechanical engineer in Newburyport. In 1882 he went to China in the employ of the Chinese Government to build a cotton mill at Shanghai, owned by Li Hung Chang, then viceroy of the province, remaining there for twenty years and doing much to introduce European methods of cotton manufacture into the country.

WILLIAM J. EDWARDS

William J. Edwards died at his home in Binghamton, N. Y., on April 21, 1912. Mr. Edwards, who was born in Albany on February 20, 1873, received his preparatory education at the Centenary Collegiate Institute at Hackettstown, N. J., and his technical training at Pratt Institute in Brooklyn. Upon his graduation in 1893 he entered the employ of the Tide Water Oil Company in Bayonne, N. J., where he became acting master mechanic of the company, with charge of the design and construction of the complete oil refinery, and later superintendent of the can and case department. He was also during his residence in Bayonne president of the Bayonne Castings Company, which made a specialty of monel metal castings.

In 1911 he removed to Binghamton, N. Y., to become vice-president and treasurer of the Binghamton Clothing Company, a position which he held at the time of his death.

ELMER S. FARWELL

Elmer S. Farwell who died in Kansas City, Mo., on December 13, 1912, was born December 20, 1863, at Morrison, Ill. He received his education at the Baldwin University, Baldwin, Kan., and later at the Rensselaer Polytechnic Institute, Troy, N. Y., being graduated in the class of 1891. For a time he was in the employ of the Illinois Steel Company at South Chicago, going from there to the Patent Office at Washington, where he was assistant examiner in the Division of Steam Engineering. During this period he was also an instructor in the department of steam engineering of Columbian University, Washington, D. C. In 1898 he came to New York to take up work with the International Paper Company, but later retired from their employ to open an office for himself as consulting engineer. In this capacity he was retained by the Yellow Pine Paper Mill Company to install their evaporator system and later to rebuild their plant. In the fall of 1911 he moved to Kansas City, where he continued his consulting practice until shortly before his death.

LEE D. FISHER

Lee D. Fisher was born December 22, 1875, at Elyria, Ohio, and died February 8, 1912, at his home in Joliet, Ill. In 1897 he was graduated from Washington University as an electrical and mechanical engineer and accepted a position as boiler and elevator inspector for the Union Casualty and Surety Company of St. Louis. Four months later he was transferred to the company's New York office. When the Spanish-American War broke out he was appointed chief assistant engineer on the flagship New York and later was transferred to the transport Buffalo in charge of the engine room, making a trip to Manila, where he was temporarily assigned to shore duty as engineer in charge of fortifications during the insurrection of Aguinaldo. He returned to the United States on the Buffalo as chief engineer.

Mr. Fisher joined his father in 1900 in electric railroad building out of Columbus, Ohio, serving as chief engineer in the construction of three roads radiating from that city. In 1903 he aided in building the Joliet, Plainfield and Aurora Railroad and the Joliet and Southern Traction Company's line. In 1911 he accepted service with the Public Service Company of Northern Illinois, having charge of franchise matters for the company until last October, when he was

made assistant to the vice-president and placed in charge of the publicity department of that corporation.

ERWIN GRAVES

Erwin Graves was born near Tiffin, Ohio, August 19, 1850, and was educated in the public schools. At the outset of his career he was for several years surveyor for railroads. He then entered the employ of J. D. Cook of Toledo, Ohio, where he assisted in the building of the Sandusky Water Works. During his four years' residence in Sandusky he was superintendent of water works and city engineer. From Sandusky Mr. Graves went to Quincy, Ill., where he built the water works, later going to Atlantic City, N. J., in the employ of R. D. Wood & Company, for the same purpose. In 1885 he became connected with the Camden Iron Works, Camden, N. J., and was until 1890 assistant superintendent. In 1894 he returned to the works as superintendent, a position which he held at the time of his death on December 13, 1912.

Mr. Graves was a member of the Engineers Club of Philadelphia for many years, but resigned from it before his death.

JOHN T. HAWKINS

John T. Hawkins was born in Abingdon, England, December 22, 1828. He came to America when eight years old and received his education in the public schools of New York City. His first employment was with Hoe & Company, where he remained until July 1861, when he joined the Navy as third assistant engineer. For efficiency of service he was promoted to a chief engineer's berth, in which capacity he served until the close of the war. He was present at the attacks on Forts Jackson and St. Phillips, the capture of New Orleans and many less important engagements. The following four years he taught engineering chemistry and physics at the United States Naval Academy at Annapolis, and then together with George R. Holt, another naval officer, resigned from the service and engaged in spool making at Salisbury, Vt. His next connection was with the Campbell Printing Press and Manufacturing Company, Taunton, Mass., of which he became president and was in entire charge of the mechanical part of the business. During this period he made many improvements on printing machinery and had taken out over fifty patents on his inventions. In 1897 he became consulting engineer for the Crown Cork & Seal Company, Baltimore, Md., and three years later retired from active business. He died April 29, 1912.

MILTON P. HIGGINS

Milton P. Higgins, for many years superintendent of the Washburn Shops of Worcester Polytechnic Institute, and widely recognized as the originator of this plan of education as well as for his ability in the industrial and commercial world, died at Worcester, Mass., on March 8, 1912.

Mr. Higgins was born December 7, 1842, in Standish, Me., and early in his boyhood showed his liking for machinery. At the age of seventeen he entered the shops of the Amoskeag Manufacturing Company of Manchester, N. H., and while devoting himself ardently to his work, managed to apply himself to study in the evening and finally to enter and to graduate from Dartmouth College, wholly through his own efforts. After graduation he became a draftsman at the Washburn & Moen Wire Works in Worcester, under the late Charles Hill Morgan, with whom he was associated later for many years at the Worcester Polytechnic Institute.

It was at this time that the plan of Mr. Ichabod Washburn for a shop that would train the students and at the same time be a commercial success was put into practice, and Mr. Higgins was selected by Mr. Morgan and other trustees of the Worcester Polytechnic Institute to carry it out. Industrial educators everywhere and those interested in such matters know how well Mr. Higgins succeeded, building up several lines of business for which he himself designed the machinery, such as machine tools, special grinding apparatus, and the hydraulic elevator of the direct-acting plunger type, and the work of its graduates has been a constant testimony to the thoroughness of administration during the twenty-eight years he was superintendent.

Mr. Higgins' active mind had time for other projects and in 1885 he organized, with George I. Alden, also of Worcester Polytechnic, the Norton Emery Wheel Company, the personnel of which has been altered only by Mr. Higgins' death. The development of the great abrasive industry of the present Norton companies is due largely to the foresight and judgment of Mr. Higgins.

In 1889 Mr. Higgins was granted a year's leave of absence from the Washburn Shops, so that he might organize the shops of the Georgia School of Technology started at that time in Atlanta, Ga., and which is still in successful progress. He was also called upon to plan the shops of the Miller Manual Labor School of Virginia.

In 1896 Mr. Higgins resigned his connection with the Worcester Polytechnic Institute, although through his appointment later on the

Board of Trustees he still maintained his interest in it, and organized the Plunger Elevator Company, which had its inception in the work of the school. Mr. Higgins' genius for organization also showed itself in his formation of the Worcester Pressed Steel Company in 1904.

At the time of his death, in addition to his association with the companies already described, Mr. Higgins was president of the Manchester Supply Company of Manchester, N. H., and of the Sanford Riley Stoker Company of Providence, R. I. He was a director of the Mechanics National Bank of Worcester, and was president of the Board of Trustees of Worcester Polytechnic Institute. He was a charter member of the Society for the Promotion of Industrial Education.

Mr. Higgins was a charter member of this Society and served as one of its vice-presidents from 1901 to 1903. In 1899 he presented before the Society two papers on the Education of Machinists, Foremen and Mechanical Engineers, giving his educational theories, which created wide interest. The Society was represented at Mr. Higgins' funeral by Honorary Vice-Presidents G. I. Rockwood, A. L. Smith, H. B. Smith, C. A. Read, Geo. I. Alden and Paul B. Morgan.

ALBERT W. JACOBI

Albert W. Jacobi was born August 24, 1861, in Newark, N. J. His professional service began in 1878 when he started as an apprentice in the machine trade at the Hewes and Phillips Iron Works of that city. His first two and a half years were spent on the Porter-Allen engines which were at that time built at these works. When Mr. Porter removed to Philadelphia he was placed in charge of the construction of two small engines, and assisted in the erection of numerous machine tools. He then assisted in the drawing room in which department his apprenticeship was completed. During his stay here he realized that his technical education had been much neglected, and therefore decided to attend the Ohio State University. While at college he assisted in designing and constructing the steam heating system in the chemical laboratory as well as in other minor constructions. Upon graduation in 1884 he was re-engaged by his former employers, his attention now being confined to machine tools, many of which were of heavy type, to steam engines, boilers and special machinery. He later became identified with the American Lead Pencil Company of Hoboken, as superintendent, but was obliged to give up this position on account of ill health. In 1901 he became connected with the Gould and Eberhardt Machine Tool

Company of Newark, as chief engineer, but three years later relinquished this for a position as superintendent of the Automobile Club of America. In 1908 he entered into private practice as a consulting engineer, in which capacity he was engaged until the time of his death, February 18, 1912.

JAMES P. S. LAWRANCE

James P. S. Lawrance was born in Philadelphia in 1852 and attended the Episcopal Academy in that city. He was matriculated at the Lehigh University and graduated in 1873 with the degree of M. E. He served about one year under instruction in the machine shop of John Roach & Sons, Chester, Pa., to qualify for the Engineering Corps of the Navy, which service he entered as second assistant engineer in March 1875. He made a three years' cruise on the Asiatic Station and another on the Pacific Station, also a six months' cruise on the North Atlantic Station in a sea-going monitor. In April 1883 he was ordered to duty in the office of Naval Intelligence in the Bureau of Navigation, Navy Department, and in June 1883 was commissioned a past assistant engineer in the United States Navy. Subsequently Captain Lawrance served at the Norfolk and the Washington Navy Yards; the Homestead Steel Works and the Thurlow Steel Works. He took part in the "battles" of Cardinas and Manzanello during the war with Spain and made voyages through the Straits of Magellan, the Suez Canal, up the Amazon River about 2000 miles, and around the world. He was promoted in the regular course up to the rank of chief engineer in the Navy and by virtue of the act of Congress approved March 3, 1899, was transferred into the line of the Navy and promoted to the rank of commander. At his own request his name was transferred to the retired list June 30, 1905, which promotion carried with it the rank of captain. He died January 16, 1912.

Captain Lawrance was a member of the American Association for the Advancement of Science.

FREDERICK J. MAYER

Frederick J. Mayer was born in Bremerhaven, Germany, July 4, 1853. He received his technical education at the University of Stuttgart, and came to this country in 1870, after having seen active service in the Franco-Prussian War. He engaged with the Bartlett Hayward Company of Baltimore, Md., as draftsman,

and while with them, as chief engineer, he designed, erected and operated many of the large gas plants constructed by the company. Among them may be mentioned the Consumers' Gas Fuel and Light Company of Chicago, in 1883, and the Nassau Gas Company's water gas plant, Brooklyn, in 1891. In 1906 Mr. Mayer severed his connection with the Bartlett Hayward Company and organized and established the Didier-March Company in the United States, taking the position of general manager.

Having made a life study of the carbonization of coal for metallurgical and domestic use, to him must be credited the establishing of some of the best foreign apparatus for such purposes in the United States by both the intermittent and continuous system in vertical retorts for gas-house practice. He not only secured these rights for the United States, but promoted the companies, erected the plants and put them into successful operation. Mr. Mayer had invented many improvements for the manufacture of illuminating gas, patents of which were secured in the United States, Germany, France, and England.

At the time of his death, September 4, 1912, he was still an ardent and tireless worker, well known in this country and abroad. He was a member of the American Gas Institute, the Illuminating Engineering Society, Chemists Club, American Peat Society and many other technical organizations.

GEORGE WALLACE MELVILLE

A career which has added luster to the entire engineering profession was brought to a close on March 17, 1912, in the death of Rear-Admiral George Wallace Melville, U. S. N. Retired, at his home in Philadelphia, Pa., after a brief illness.

Born of Scotch parentage illustrious for its bravery, George Wallace Melville was one of the first to volunteer in his country's service at the outbreak of the Civil War. In 1804, James Melville of Sterling, Scotland, had come to New York, bringing with him his family, one of whom, Alexander Melville, a chemist, became the father of the noted engineer. George Wallace, oldest son of Alexander and Sarah Douther Wallace Melville, was born in New York on January 10, 1841, educated in the public schools, and showing early mechanical ability, sent to the Polytechnic School of Brooklyn and, later, to a religious academy noted for its excellent mathematical training. He had entered the engineering works of James

Binn in East Brooklyn, and had acquired the foundation for that practical skill which served him so well in many a later event, when war was declared.

Melville was at once appointed an officer of the Engineering Corps of the United States Navy, for which he was well fitted despite his extreme youth, and throughout the war saw almost continuous service. He will perhaps be best remembered for his daring work in connection with the capture of the Confederate cruiser Florida in the neutral waters of Brazil—a breach of international law, but a justifiable and clever strategy.

The war ended, Melville's adventurous spirit soon sought for new foes to conquer, and when in 1873 the sealing steamer Tigress was chartered to rescue the remnant of Captain Hall's last and fatal voyage to the Arctic, Melville volunteered as her engineer officer. The White North proved an antagonist worthy of Melville's bravery and skill, and it was for his share in the long struggle with the enemy, conquered only within the past two years, that his name first became honored throughout the civilized world. When in 1879 the Jeannette, commanded by Lieut. George W. De Long, embarked upon an expedition to the North Pole through Bering Strait, hitherto an untried route, Melville was of the staff, although the department reluctantly assented to his assignment because of the difficulty of filling his place. On the voyage he was called by De Long "a tower of strength," "always self-helpful and reliant in emergency," "of indomitable energy," yet ever cheerful and with a smile and word of encouragement for all. "When he does retire," says De Long, of the time of crisis, "he lies awake, planning some new means of pumping a ship by steam which will be more economical than the main boilers."

Such planning was much needed, for soon after leaving California the vessel was caught in the ice and never after escaped from the floe which held her. In 1881, Melville manned a small force sent by De Long to plant the American flag on what is now known as Henrietta Island, which proved a most hazardous undertaking and which has been most graphically described by Melville himself in his book, "In the Lena Delta." After a three days' journey of constant danger and struggle with the moving ice, hindered by various mishaps and by the frightened dogs, the island was triumphantly claimed for the American government. Of this feat De Long says: "If this persistence in landing upon this island, in spite of the superhuman difficulties he encountered, is not reckoned a brave and meritorious

action, it will not be from any failure on my part to make it known."

On June 7, 1881, the Jeannette, worn out with the crushing force of the moving ice, went down, leaving her crew upon the floe, the nearest source of food 500 miles away in the Lena Delta, and the nearest known land, the desert New Siberian Islands, 150 miles distant. For forty-one days they toiled on, Melville ever ready to aid the sick and discouraged and full of energy and expedients. At length coming upon an island hitherto unknown, now called Bennett Island, they rested for nine days and then when the Arctic winter was setting in, embarked southward in three boats, commanded respectively by De Long, Chipp and Melville. Separated by a terrific storm, Melville's crew were finally brought to port, alive but in a pitiable condition. Chipp's vessel had gone down in their sight, a victim of the waves; De Long's fate they could only conjecture. Instead of pushing his way southward, "whither," as Bennett says in his, The Steam Navy of the United States, "a warmer sun was shining and where lay the great Russian road leading to the borders of civilization, to the railway, to the blue sea, to home," Melville set out to find De Long, who in his own heart he feared had perished. "Few men," says Bennett, "after passing through all the perils that Melville had survived, would have thrown away that seeming last chance of personal safety, but he, being cast in rare, heroic mould, turned his back upon the means of saving himself and set his rugged face to the northward, forcing his way into the darkness and the awful silence of the Arctic winter like one deliberately invading the dominions of death."

How Melville pressed forward, weak and footsore, suffering from cold and exposure, quelling the mutinies of his native companions from time to time, only to fail in his quest; how, when spring came, he went forth undaunted a second time, to discover at last the bodies of the brave De Long and his party and to bury them, as he has himself written, "in the sight of the spot where they fell, the scene of their suffering and heroic endeavor, where the everlasting snows would be their winding sheet, and the fierce polar blasts would wail their wild dirge through all time,"—is now a matter of common knowledge and need not here be more fully recounted.

In the spring of 1884 Melville made his last trip to the North, this time with the Greeley relief expedition, and was one of the first to reach the dying men at Cape Sabine, a fit ending to this chapter of his career.

The Government had not been unmindful of Melville's broad knowledge of engineering and his intrepid spirit, and in 1887 when the new Navy was in process of building, Admiral Melville was appointed Engineer-in-Chief with the relative rank of Commodore.

At this time the four Roach Cruisers had been completed and contracts had been placed for a few of the succeeding vessels. Except for these few, however, the machinery of the new Navy was all designed by Melville or under his supervision until the time of his retirement sixteen years later. Even this does not represent the total amount of designing done by the Bureau under his direction, since the work was so radically new that tentative designs had to be prepared in many cases, which subsequent changes in the general features of the vessel caused to be given up.

Machinery aggregating more than 1,000,000 h.p. was designed and built under his supervision for vessels of all types from battleships to torpedo boats, involving millions of dollars. He was the first engineer-in-chief to advocate and to actually install water tube boilers. The first to be thus fitted out was the Monterey, equipped with Ward boilers of about 5,000 h.p. He also adopted water tube boilers exclusively for sea launches with the best results, but did not allow himself to be carried away unduly with the idea. He felt that the best interests of the Navy required that the first boilers installed be tested thoroughly before adopting this type as a settled policy.

Ever vigilant to keep our Navy in the forefront of progress he consistently demanded the highest possible speed in vessels of all types. In 1898, when other nations were building large battleships of 18 knots speed or over, it was proposed by the Navy Department to construct new vessels copied after the slower 16-knot battleships of previous years; but the Admiral's determined stand and the willingness of contractors to furnish ships capable of attaining the higher speed enabled us to get vessels of 18 knots. As another evidence of this policy, reference should be made to the two triple-screw flyers, Columbia and Minneapolis, which were the first very large vessels to be equipped with triple screws, each vessel developing about 20,000 h.p. and breaking the speed record for naval vessels larger than torpedo-boats at that time. This was one of the boldest pieces of designing which has been done in the Navy Department and is but another evidence of the courage and foresight displayed by Melville upon all occasions where progressive action was demanded.

He applied the method of testing the speed of vessels now known as the "Standardized Screw," and at the time of his retirement had completed an extensive investigation of the use of fuel oil on shipboard.

In the war with Spain, bitterly opposed by Melville with all his customary strength of opinion, his foresight and determination were nevertheless responsible for the equipment of Key West as a naval base and the sending out of the Vulcan, the first repair ship to be used in warfare.

In 1899 came the culmination of the long fight which the engineers in the navy had made to secure actual rank as naval officers instead of relative rank with professional titles. Inasmuch as their work has come to be largely executive, involving the command of a considerable portion of the ship's company, the engineers felt that they should have a title commensurate with their duties. This was opposed for years by the line officers, but the splendid work of the Engineer Corps under Melville and his own personality brought about a better feeling and the Personnel Board was appointed, presided over by Assistant Secretary of the Navy Roosevelt. The result of the deliberations of this board was a reorganization of the entire personnel of the navy, which consolidated the engineers with the line. It cannot be doubted that the work and bearing of Melville and the accomplishments of his department were largely responsible for this sweeping change.

In 1903 Admiral Melville retired from the service in accordance with the sixty-two year limit, though still active and vigorous both mentally and physically, and several years later opened a consulting office in Philadelphia in partnership with J. H. Macalpine, the joint inventor with him of the steam turbine reduction gear for ship and other purposes, which has been described in their book upon the subject, "Broadening the Field of the Marine Steam Turbine," published in 1909. Just before the close of his administration an elaborate series of tests on the use of fuel oil was made under his direction by the U. S. Naval "Liquid Fuel" board, and published as a public document.

Throughout his life Melville's inventive genius was constantly in evidence and there is a long list of inventions to his credit during his work as an ordnance engineer which were never patented. In later years he made numerous patented inventions.

Admiral Melville was justly the recipient of many honors. On

his way home from the Jeannette expedition he was granted a private audience by the Czar and Czarina of Russia and was made a Knight of St. Stanislaus of the First Class. Congress in 1900 conferred upon him a gold medal for his heroic service in the Arctic regions and advanced him a grade. He was an honorary member of the Royal Swedish Society of Anthropology and Geography, a member of the National Geographical Society of the United States, the Geographical Society of Philadelphia, the Philosophical Society, the Washington Academy, the Naval Architects and Marine Engineers, and an honorary member of Franklin Institute and of the American Society of Civil Engineers. He was subsequently made also an honorary member of the Institution of Naval Architects of Great Britain, and honorary degrees were conferred upon him by a number of colleges. His service in the Civil War made him a member of the Grand Army of the Republic, of the Naval Order of the United States, and of the Military Order of the Loyal Legion, composed of commissioned officers of the war. By the latter he was shown much honor and was at the time of his death Commander of the Pennsylvania commandery and Commander-in-Chief of the Order. Some years ago the Loyal Legion placed a fine bust of Admiral Melville, done in bronze, in the Philadelphia Museum.

Admiral Melville became a member of the Society in 1893 and was elected to honorary membership at the Spring Meeting of 1910, held at Atlantic City, where he delivered an address on The Engineer's Duty as a Citizen. From 1895 to 1897 he was a vice-president of the Society and in 1898 became its president, filling the office with much distinction, and giving an interesting account of Engineering in the United States Navy, its Personnel and Material, as his presidential address at the Annual Meeting of 1899. When the Society held its meeting in Washington, D. C., in the spring of 1909, one of its prominent features was an address on the Engineer in the Navy delivered by Admiral Melville, followed by the presentation of his portrait, painted by Sigismond de Ivanowski, to the National Gallery. On the occasion of his seventieth birthday, a number of Admiral Melville's friends presented him with a silver platter of fine workmanship and a memorial resolution, handsomely engrossed, which now hang in the rooms of the Society. The Society was represented at Admiral Melville's funeral on March 20, 1912, by Honorary Vice-Presidents Walter M. McFarland, James M. Dodge, Past-President, Coleman Sellers, Jr., and Theo. N. Ely.

SAMUEL TENNEY MUDGE

Samuel Tenney Mudge died at his home in South Manchester, Conn., on September 8, 1912. Mr. Mudge was born in Brooklyn, N. Y., on February 22, 1883, and received his education at the Stevens Institute of Technology, from which he was graduated with the degree of M.E. in 1906. He entered the service of the American Sugar Refining Company in Jersey City in the same year, and two years later became instructor in machine design in the University of Michigan. At the time of his death he was general manager and superintendent of the mills of the Rogers Paper Manufacturing Company, South Manchester, Conn.

EDWARD S. RENWICK

Edward S. Renwick, who died March 19, 1912, was born January 3, 1823, in New York, and when thirteen years of age entered Columbia University, the youngest member of his class. After graduation in 1839 he engaged in the manufacture of iron with the New Jersey Iron Company at Boonton, as assistant and bookkeeper to the superintendent. In 1844 he was employed to examine and report upon some mines in Maryland and was afterward sent to England to attend to matters appertaining to them. Here he had the opportunity to examine the best iron works in both England and Wales. In the fall of 1845 he returned to America and became superintendent of the Wyoming Iron Works at Wilkes-Barre, Pa., which comprised a merchant mill for rolling bars, a sheet mill, a mill for rods and hoops and a nail factory. He also put up a small blast furnace in the same town and engaged in the manufacture of pig iron. In the spring of 1849 Mr. Renwick went to Washington and associated himself with Peter H. Watson, subsequently Assistant Secretary of War under Stanton during the Lincoln adminstration, in the business of solicitor of patents and expert in patent causes in the United States Courts. While at Washington he made several inventions, the most important of which was the original self-binding reaping machine, which cut the grain, gathered it into gavels, compressed it, and bound it into sheaves, with wire, but preferably with twine. It remained dormant, however, until the seventies, after the expiration of the patents, when the impossibility of obtaining sufficient laborers to harvest grain in the old way forced it into use. This invention, with

small improvements, has yielded millions to the makers of reaping machines, and has added untold wealth to the country.

In December 1854 Mr. Renwick returned to New York, and his first employment was as consulting engineer to Harrison Gray Dyer, who had become president pro tem of the New Haven Railroad. His employment in this position terminated with the resignation of Mr. Dyer and he resumed the practice of his profession as patent expert and consulting engineer. The most important engineering matter in which he was called upon to act was the repair of the Great Eastern steamer while afloat. This consisted in covering with iron plating a fracture of the bilge 27 feet beneath the water, 88 feet long and 10 feet 6 inches broad at its widest part, a feat declared impossible by other experts. He did the work in conjunction with his brother, Henry B. Renwick.

From 1865 Mr. Renwick's employment as expert in patent causes was continuous. He had probably been subjected to the longest cross-examinations of any expert. Thus in one of his early cases his cross-examination lasted twenty-one days, with the result that the United States Circuit Court and afterward the United States Supreme Court on appeal, adopted the construction of the patent given by him in favor of the defendant. In another case the cross-examination by the complainant's counsel was carried on in sessions of from one to three days separated by intervals, the whole amounting to twenty-two days running through six months, when the case was compromised for $15. In the Leffel waterwheel case he was cross-examined thirty-five days, the decision of the court being in favor of the plaintiff on whose behalf he had been called as expert.

Mr. Renwick was greatly interested in artificial incubation, and it was due largely to his efforts and inventions that the raising of young chickens was rendered a paying industry. He was also responsible for many other inventions, among which are the balanced compound steam engine, and the system of surrounding that portion of the shaft of a twin propeller which extends beyond the vessel with a casing of sufficient size to permit this portion of the shaft to be inspected to the stern bearing. Both of these are described in his English patent of 1868. The former invention did not come into use until the last decade of the last century; and the latter until the accident to one of the shafts of the City of Paris, since which it has been employed in all the great sea-going steamers with twin screws. In 1893 Mr. Renwick published his work on Patentable Inventions.

He was a member of the American Chemical Society, the Engineers Club, the Union Club, the New York Yacht Club, the St. Nicholas Society and other similar organizations.

HENRY S. ROBINSON

Henry S. Robinson was born at Meredith, N. H., January 22, 1831, and died April 4, 1912. After completing his academic education, he took up the active duties of life in 1847 as assistant engineer of the Hamilton Manufacturing Company at Lowell, Mass., and later became consulting engineer for the same company. In 1849 he went to the York Mills at Saco, Maine, as chief engineer, where he remained six years. From 1855 until 1861 he and his brother, James R. Robinson, were in business at Clinton, Mass., as consulting steam engineers. During the Civil War he served for a short time as assistant engineer in the Navy and later in the Army as lieutenant in the 36th Regiment of Massachusetts. From 1865 until 1873 he had charge of the steam plant at the Pacific Mills, Lawrence, Mass., and in 1873 engaged in the business of boiler manufacturing at Boston, at the same time acting as consulting engineer for many of his former clients. In 1892 he retired from active business and allied himself with The Atlantic Works, Boston, of which company he was vice-president from 1892 until 1898, and president from 1898 until his death.

From an engineering standpoint, Mr. Robinson's life was particularly noteworthy for the practical results which he achieved in improving methods of steam boiler construction and operation, and in convincing his clients and customers of the desirability on economic grounds, and duty for humanitarian reasons, to design, construct and operate steam boilers with a careful and intelligent regard for the natural laws affecting them. Before his death he had the satisfaction of knowing that methods which he had advocated and practised many years before were being quite generally adopted by engineers and boiler manufacturers, and in some particulars were being required by legislative regulations.

J. M. ROBINSON

J. M. Robinson, president of the Crex Carpet Company of New York, who died on March 30, 1912, was born in New York on July 19, 1849, and educated in the public schools. He received his engineering training in Cooper Institute and through practical experience.

Mr. Robinson had charge for some time of the linesmen and building of the Western Union Telegraph Company, and was associated with the Mills Estate, erecting the Mills Buildings

in New York and San Francisco, and having active management of the property. He was also interested in many commercial enterprises, including the company of which he was president at the time of his death.

Mr. Robinson was a resident of Teaneck, N. J., and actively interested in the Board of Education and in political and social affairs of that township.

HARRY M. ROSE

Harry M. Rose was born at Albany, N. Y., November 22, 1856, and died at Elizabeth, N. J., December 25, 1912. He received his education in private schools and under private tutors, and afterwards entered the service of the F. O. Matthiessen and Wiechers Sugar Refinery, with a view to learning the business. Upon completion of a four years' course he went to San Domingo and there engaged in sugar harvest work. Following this he went to the Philippine Islands for the George L. Squier Manufacturing Company of Buffalo, N. Y., and was engaged in that country for a period of three years in installing the machinery on sugar and rice plantations. Upon his return from the Philippines he made a number of trips to Mexico and Central America as constructing and consulting engineer in the line of plantation machinery.

AMORY E. ROWLAND

Amory E. Rowland was born July 2, 1852, in Brooklyn, N. Y., and educated in the Sheffield Scientific School of Yale University. He gained experience after graduation in practical mechanics in the Continental Iron Works in Brooklyn and had been connected with the firm of F. C. & A. E. Rowland in New Haven and its successor, the Rowland Machine Company, about thirty-five years. At the time of his death on May 7, 1912, he was secretary and treasurer of the company.

Mr. Rowland was a trustee of the New Haven Savings Bank, director and prominently identified with the New Haven Colony Historical Society, the Graduates Club of New Haven, the Organized Charities Association, and other similar organizations.

GEORGE H. SCHULTE

George H. Schulte was born in Cincinnati, Ohio, April 16, 1848, and at the age of sixteen went to Little Rock, Ark., where

he was employed in the Fones Brothers' hardware store. He became its manager in 1870. Twelve years later he went to Milwaukee, Wis., where for the next twenty years he was superintendent of the Milwaukee Harvester Company, having full charge of the mechanical part of the works. Under his management the harvester was brought to so high a degree of perfection that it found a world-wide market. The company was purchased by the International Harvester Company of America in 1904, and Mr. Schulte remained as vice-president and general manager for one year, when he was compelled to retire on account of ill health. From 1906 to 1908 he was director and general manager of the J. I. Case Threshing Machine Company of Racine, Wis. He died July 29, 1912.

ARTHUR C. SCOTT

Arthur C. Scott was born January 27, 1870 at Northbridge, Mass. He moved to Worcester when eleven years old and obtained his education in the public high school there and the Worcester Polytechnic Institute. His first employment was in the freight office of the New York, Providence & Boston Railroad and two years later he went to work for the Draper Machine Tool Company of Worcester where he remained for seven years. In 1903 Mr. Scott entered the employ of the Norton Company as traveling salesman, and in 1909 was sent to Great Britain to sell grinding wheels and to investigate conditions abroad. He died November 22, 1912.

JAMES B. SINER

James B. Siner, a mill engineer of Boston, died on September 17, 1912, at his home in Malden. Mr. Siner who was born in Lowell, Mass., on April 3, 1835, was educated in the public schools of that city and served his apprenticeship in the Lawrence Machine Shop. He began his career as mechanical superintendent of the Washington Mills at Lawrence, with which he was associated for twenty-two years, and was at the time of his death in charge of the appraisal department of the Factory Mutual Fire Insurance Company of Boston.

HENRY W. SPANGLER

Henry W. Spangler was born at Carlisle, Pa., January 18, 1858. He was graduated with high rank at the United States Naval Acad-

emy in the class with two others who have attained to prominent positions in the field of engineering education, namely, Ira N. Hollis, head of the engineering department at Harvard University, and M. E. Cooley, dean of the department of engineering at the University of Michigan.

Professor Spangler was assistant engineer in the United States Navy from 1878 to 1889, although for about half of that period he was connected on detached service with the faculty of the University of Pennsylvania, first as assistant professor of mechanical engineering from 1881 to 1884, and from 1887 to 1889, and then as full professor, holding the Whitney professorship of dynamical engineering. During the Spanish-American War he served for a brief period as chief engineer in the United States Navy. With that exception he was in the service of the University of Pennsylvania as head of the mechanical and electrical engineering department continuously from 1887 till his death.

He was the author of a number of standard textbooks and numerous technical papers and professional reports. The textbooks from his pen embrace Valve Gears; Notes on Thermodynamics; Elements of Steam Engineering, which he wrote in collaboration with A. M. Greene and S. M. Marshall; Graphics; and Applied Thermodynamics. He was a member of the American Society of Naval Architects and Marine Engineers, the American Society of Naval Engineers, the American Society for Testing Materials, and the Society for the Promotion of Engineering Education, the Franklin Institute and the Engineers Club of Philadelphia. He was a member of the Advisory Council of the Engineering Congress of the World's Columbian Exposition in 1893 and of the Jury of Awards at the Buffalo Exposition in 1901.

In 1896 the University of Pennsylvania conferred upon him the honorary degree of Master of Science and ten years later the degree of Doctor of Science. He died March 17, 1912, at his home in Philadelphia.

Edgar Marburg, professor of civil engineering at the University of Pennsylvania, pays him the following tribute:

> As a writer, his chief characteristics were, perhaps, his painstaking efforts to present the subject in the simplest and clearest manner consistent with the intended scope of treatment, and to keep in view the practical requirements of prospective engineers, rather than theorists.
>
> As a teacher, he was lucid, stimulating, progressive, and always intensely practical. His first concern was to help the students to gain a firm grasp of the underlying principles of the subject, and then to encourage them to rely on their

own resources in the application of these principles. On no point, perhaps, was he more insistent than that of individual responsibility, which his students were required to assume in every branch of their work.

A strict and almost military disciplinarian, he was no less rigid in the standards which he applied to himself. The respect and admiration in which he was held by his students ripened into affection as they came to see him at closer range. There were few graduates who failed to turn to him at some time for helpful counsel in the perplexities of later years, or who failed to accept it, even though it ran counter to their own promptings.

He possessed to a remarkable degree the faculty of perceiving clearly, and almost intuitively, the essential elements of a seemingly difficult problem or complex situation, and he was as quick in action as in perception. Few excelled him in the clear discernment of the fallacies of an argument or in the directness of the challenge of such fallacies. Of a thoroughly progressive bent, he did not allow himself to be beguiled into strange paths by the educational fads and follies of the hour. The business of education was to him a serious business, with which liberties were not to be taken lightly.

FRANCIS H. STILLMAN

Francis H. Stillman, president of The Watson-Stillman Company, and a prominent figure in machine tool and engineering industries, died suddenly on February 16, 1912, at his residence in Brooklyn.

Mr. Stillman was born in New York on February 20, 1850, and was graduated from Yale in the class of 1874 with the degree of B.S. On leaving college, Mr. Stillman first associated himself with the Cottrell Printing Press Company, at Westerly, R. I. In 1883 he became president of the firm of Watson & Stillman which succeeded Lyons & Company, and in 1904 the firm was incorporated as The Watson-Stillman Company, Mr. Stillman remaining president up to the time of his death. Under his direction the firm early became prominent in the hydraulic engineering field and has a large and active plant in Aldene, N. J. Mr. Stillman was also president of the Bridgeport Motor Company and of the Pequannock Commercial Company, and a director of other manufacturing firms. He was one of the recognized pioneers in hydraulic machine tool construction.

Mr. Stillman was a member of the Engineers Club, the Hanover Club of Brooklyn, and the treasurer as well as a director of the National Association of Manufacturers. He organized and was the first president of the Machinery Club of New York, and was also the first president of the National Metal Trades Association.

Mr. Stillman was actively interested in the Gas Power Section of the Society, as well as in the work of its main body, and was serving as the chairman of its executive committee at the time of

his death. At a meeting of the committee on February 29, 1912, the following tribute was spread upon the minutes:

The Committee desires to enter upon its records something of their appreciation of the long and effective services of Mr. Stillman as a member of the Section and of the Executive Committee, and finally its Chairman. He died while carrying the burdens and privileges of that office.

Mr. Stillman brought to the duties of shaping the policy of the Gas Power Section of the Society, his wide experience in administrative affairs, his shrewd and clear-sighted judgment, and that genial kindness of spirit which was one of his greatest charms. In all the difficulties and problems of starting such an enterprise, his abilities and his assiduity of service were splendid factors in bringing the Gas Power Section to its present standard of usefulness and effectiveness. His associates will miss his high standard of business integrity, his keen personal interest in the duties which he had to do, and his willingness to take trouble for the things which he thought worth while. His colleagues will miss also his friendship and the pleasure of intimate association with him.

The Society was represented at Mr. Stillman's funeral by Honorary Vice-Presidents Fred R. Low and Jesse M. Smith.

GEORGE H. SULZER

George H. Sulzer, chief designer and manager of the centrifugal pump department of the Worthington Hydraulic Works, Harrison, N. J., died in New York April 20, 1912. Mr. Sulzer was born in Winterthur, Switzerland, October 21, 1877. He received his early education at the public schools and industrial college of Zurich. His professional training was obtained at the Polytechnic Institute of the same town, where he received his diploma in 1902. For several years after his graduation he was assistant in the department for centrifugal pumps and turbines at the Polytechnic Institute. In 1903 he sailed for America and found employment with R. D. Wood & Company, Camden, N. J., and later on the engineering staff of the Buffalo Forge Company. In 1906 he was engaged by the Henry Worthington Company to take charge of the design of centrifugal pumps, which had always been his specialty. He had an unusually intimate understanding of the theory and construction of rotating machinery, particularly of the newer types of pumps, and his services were most valuable in the solution of special problems, even though he was often handicapped by the prevailing commercial tendencies and conditions. He was a member of the Newark branch of the Deutscher Technischer Verein, being its president for two terms. He was also a member of the Vereinigung der Schweizerischen Techniker.

E. HARRISON SYMINGTON

E. Harrison Symington was born in Baltimore, Md., March 1, 1879, and was educated in the public schools there and in the technical training school at Bethlehem, Pa. He entered Lehigh University in the class of 1898 and was graduated with the degree of M.E. His first experience was in the shop of the Lehigh Valley Railroad, with which company he remained till he was made engineer of tests, serving in that capacity for one year. In 1897 he entered the employ of Kilborne & Jacob Manufacturing Company of Columbus, Ohio, where he was in charge of the engineering and shop practice. He resigned this position to become manager of western sales of the T. H. Symington Company, and subsequently of their foreign business. When the concern opened its new malleable foundry in Rochester, N. Y., 1909, he was made works sales manager. His last years were spent as mechanical expert for the company, especially devising improved methods for draft gear designs for freight equipment.

Mr. Symington died in Baltimore, September 5, 1912.

CHARLES N. TRUMP

Charles N. Trump was born at Newport, Bucks County, Pa., on May 30, 1829, of Quaker parentage, and received his education in a private school of Alexandria, Va., his father having engaged in the lumbering business in Baltimore. In spite of opposition by his family, who wished him to enter one of the professions, which did not then include mechanical engineering, he apprenticed himself in one of the machine shops of Baltimore.

Naturally a student, Mr. Trump absorbed all the engineering information available, and becoming interested in gas engineering, opened an office in Philadelphia for the erection of gas works, especially to develop a system of manufacturing gas from resin. He erected several works in different parts of the country, but during the Civil War was obliged to abandon his business and to take up insurance. In 1865 he was engaged by Russell, Birdsall & Ward, Port Chester, N. Y., to take charge of a section of the works.

In 1870 he and his brother opened a machine business in the same town, under the name of Trump Brothers Machine Company, and developed a line of manufacture of foot-power scroll saws, special chucks, and other light machines. Two years later they moved the business to Wilmington, Del., considerably enlarging it and taking

up the manufacture of knitting machinery. Mr. Trump retained for many years the presidency of the company, and retired only a short time before his death on December 30, at the age of 83.

JOHN BURKITT WEBB

John Burkitt Webb was born in Philadelphia, Pa., November 22, 1841, and entered the Philadelphia high school at the age of thirteen. Although kept at work in a store for some years after leaving school, he was always in pursuit of mechanical and physical problems. His father having patented a windmill, young Webb set about to determine the speed and angle of sails giving the maximum efficiency, and thereby was led to study higher mathematics. Finally he left the store and went to Bridgeton, N. J., where with Oberlin Smith, he formed a small company to make an electro-magnetic apparatus for playing organs automatically. The enterprise proved too great, however, for the means at hand, and was therefore abandoned. Mr. Webb then went to work in a pipe-mill and built a trip-hammer which ran many years and greatly expedited a part of the manufacture. At the outbreak of the war business became so dull that he sought employment for a year and a half with a medical journal in Philadelphia, meanwhile studying drawing in the evening at the Franklin Institute. In the summer of 1863, Mr. Webb again started in business at Bridgeton, in partnership with Mr. Smith, building special machine tools. In 1869, he entered the University of Michigan and after graduation was engaged as assistant in the civil engineering department. In 1871, Professor Webb accepted the charge of the Civil Engineering School, newly established in that institution. He held this position for eight years, during which time he made a trip to Europe for the inspection of scientific schools and for the purchase of apparatus, started a small astronomical and meteorological observatory for the school, delivered several courses of original lectures, partially developed his method of treating bridge stresses, and constructed various pieces of apparatus.

In 1879 he resigned his professorship to pursue a course of advanced scientific study abroad, and spent over two years attending lectures on pure mathematics, mathematical physics, logic, etc., and in experimental work in physical laboratories at Heidelberg, Göttingen, Berlin and Paris. A year was spent in Professor Helmholtz's laboratory constructing apparatus for, and working at, the then undeveloped action of an oscillating current through coils of wire and electrolytic solutions, with which considerable progress

was made. His skill with tools secured him the privilege of using the apparatus in the instrument-maker's shop of the university, and before leaving Berlin he was made a member of its Mathematical Society. Nearly a year was spent in Paris attending lectures at the Sorbonne and College de France and in examining technical schools, collections and methods there and in other parts of France.

In 1880 Professor Webb was appointed to a new chair of applied mathematics at Cornell University. During his occupancy he delivered original lectures on thermodynamics, mechanism, drawing and drawing instruments, acted as judge at the International Electrical Exhibition of 1884; invented a draft gage which was officially used at this exhibition, and an inertialess steam-engine indicator which excited much attention at the American Association meeting at Montreal, and published an exhaustive article on Belting to Connect Shafts which are not Parallel and do not Intersect.

Professor Webb was called to the position of professor of mathematics and mechanics at Stevens Institute of Technology in 1886. He retired in 1907 under the Carnegie Foundation Pension Fund, and had since that time been engaged in private experimental and consulting engineering work.

In 1888 he originated his floating dynamometer which is a most convenient and perfect device for measuring the power given out or absorbed by motors, dynamos and other machines. In 1892 he invented the viscous dynamometer, which is an absorption dynamometer depending on fluid friction, or viscosity, between rapidly moving surfaces. In 1900 he invented the dynamophone, which is a transmission dynamometer measuring the twist of a shaft carrying power, by a simple telephonic method of great accuracy and reliability.

Professor Webb was a member of the American Association for the Advancement of Science, American Mathematical Society, and the University of Michigan Round Table. He died in New York, February 17, 1912.

LAWRENCE WHITCOMB

Lawrence Whitcomb, who died in Boston, Mass., on May 18, 1912, was born May 2, 1863, at Malden, Mass., and was educated in the Boston and Roxbury public schools. In 1881 he left school without completing his course and entered the shoe business, becoming a salesman and factory superintendent. In 1890 he joined the organization of the firm of Whitcomb, Wead &

Company, real estate brokers and agents, which became one of the large real estate firms of Boston. Mr. Whitcomb was also interested in the development of the National Brake & Clutch Company and its successor the Cork Insert Company, which owned patents covering the use of cork for friction purposes in brake shoes, clutches, pulleys, etc., and finding that the use for these devices was very large he made a careful study of mechanical conditions and requirements and patented several new applications of his own. He was at the time of his death treasurer and general manager of this company.

MAUNSEL WHITE

Maunsel White, for many years engineer of tests and metallurgist of the Bethlehem Iron and Steel Works, died in New Orleans on October 22, 1912. He was born in Plaquemines parish, La., on March 15, 1856, of a prominent southern family. After a few years of study under private tutors and in Georgetown University, near Washington, D. C., he entered the school of technology at Worcester, Mass., and later the workshops of the Lehigh Valley Railroad at Weatherly, Pa. Here he gained a practical knowledge of mechanics which determined him to secure an engineering education. In 1875, accordingly, he became a student at Stevens Institute of Technology, from which he was graduated four years later as valedictorian of his class.

Immediately afterward, White was engaged by the Bethlehem Iron Company, under John Fritz, then largely manufacturing merchant steel and rails, now the Bethlehem Steel Company, and later became chief of the metallurgical department, taking high rank as an authority upon metals. He represented his company and was in charge of their exhibit at the World's Fair in Chicago in 1893, as well as at the Paris Exhibition of 1900, receiving personally at the latter a bronze medal of merit as a result of his company's exhibit; and he made frequent visits to Europe in the same capacity during his connection with the company, arranging among other things for the sale of great quantities of armor plate to the government of Russia, through personal interviews with the ministers of the Czar and at least one audience with the Czar himself.

Both the chemistry of iron and steel and the physical properties of these metals have been so thoroughly studied during the

past 30 years and practice concerning them has become so crystallized, that the engineer of today finds it difficult to appreciate that facts which now seem elementary and commonplace were unknown in the early days of the industry. The life of the metallurgist in those days was intense and exciting. Problems were crowded on him upon which depended the very life and existence of his company, and in the solution of these problems, in wringing these secrets from nature, Maunsel White had a large though often inconspicuous part. The confidence inspired by his work is shown in the trust placed in him by the government officials detailed to supervise the manufacture of armor plate and ordnance at the time the Bethlehem Steel Company was so largely engaged in its construction. White's O.K. on a piece of steel was regarded as final.

Maunsel White will best be remembered through his part in the extended investigation which led to the discovery of the Taylor-White process of treating steel and the development of "high-speed" tools, which have revolutionized the machine shop practice of the world. In general he has been accorded the credit which belongs to him in the discovery of this process, but few have appreciated the detailed thoroughness of his work, that true scientific devotion which enabled him to plod along, month after month, developing methodically one fact after another, and the rare ability with which he was able, in the end, to coördinate and group these facts so as to indicate the conclusions and theories towards which they tended. High-speed tools have in many cases doubled and trebled the output of metal-cutting machines, and practically all makers of high-speed steel have today adopted for their best brand, steel of almost the exact chemical composition recommended by White as a result of his experiments.

While Maunsel White will ever be remembered for his metallurgical discoveries, his friends knew him as a man of rare versatility and catholicity of taste, with an intense love of literature and the arts and marked ability in these directions as well as in the realm of applied science. He was both a poet and a mathematician, a combination in the same individual so rare that in this respect he was a marvel to all his friends. Alive to beauty, he could at the same time get at the kernel of a problem with quickness, clearness and accuracy. His work at the Bethlehem Steel Company gained for him the friendship and admira-

tion of John Fritz, the dean of the steel industry in America.

Mr. White was widely known both in America and abroad, and was a member of the Iron and Steel Institute of Great Britain and the American Institute of Mining Engineers.

I. CHESTER G. WILKINS

I. Chester G. Wilkins was born in Whitehall, N. Y., September 8, 1871. He received his early education in the public schools of Whitehall and in 1893 was graduated from Cornell University with the degree of M. E. His first work was in the railroad shops of Whitehall, and in August 1894 he went to New York entering the employ of Evans, Almirall & Company, heating and ventilating engineers and contractors, with whom he remained until May 1895. He then obtained a position with C. O. Brown, consulting engineer, Brooklyn, N. Y., and from March 1896 to May 1899 designed heating and ventilating apparatus for public school buildings. He went to Baltimore, Md., having secured a position with Henry Adams, consulting engineer, and two years later returned to New York to fill a position with Thompson-Starrett Company, with which concern he remained till his death, April 20, 1912. His work during this period consisted in the design, specification and general supervision of installations of mechanical equipment for such buildings as James McCreery & Company's store on 34th Street, New York; the Crescent Athletic Club, Brooklyn, the Adelphia Theater, Philadelphia; the Title Guarantee & Trust Company's bank and office building, Brooklyn; the New York Steam Company's boiler house; and John D. Rockefeller's residence, Pocantico Hills, N. Y.

FREDERICK W. WOLF

Frederick W. Wolf, who died in Chicago, Ill., February 17, 1912, was born in Duehren, Germany, November 27, 1837. His early education was obtained in the high schools of Weinheim, near Heidelberg, but before being able to enter a technical college he was obliged to take up some specific trade, and in 1852 entered a locksmith shop in Nuremberg, Bavaria, to serve there as an apprentice. Later he went to Switzerland, where he worked in a machine shop, returning to Berlin in order to enter the employ of a sugar machinery manufacturer. At the age of twenty-one he was sent to erect a beet sugar factory 300 miles southwest of Moscow, where he encountered a great many difficulties, being obliged to

employ men who could speak only the Russian language. After successfully completing this installation he pursued the work of erecting machinery in different German cities, until he was able to enter the technical college at Carlsruhe in Baden, where he studied mechanical engineering for four years, his last year there being in the capacity of assistant to the professor. After completing his education he was employed by the Guppy Machine Shop, an English firm in Naples, Italy, to design engines for the Italian warships. In 1866 he came to New York and obtained employment as a draftsman in a number of different firms. The next year he went west and for a short period was master mechanic for the Pere Marquette Railroad Company, at Marquette, Mich. From 1867 to 1875 he was in business as consulting engineer and architect in Chicago. Realizing the immensity of the future in artificial refrigerating machinery, Mr. Wolf purchased in 1875 the right to manufacture the celebrated Linde ice machine, patented by Prof. Carl Linde of the University of Munich. The first Linde machine erected in the United States was imported by Mr. Wolf in October 1882, and installed in the Wacker & Birk Brewery, Chicago, where it still remains. Another machine was imported and erected in the Schlitz Brewery at Milwaukee, but on account of the many difficulties surrounding the importing of these machines, Mr. Wolf decided to begin their manufacture in this country. In 1887 the Fred. W. Wolf Company, of which Mr. Wolf was president until his death, was incorporated and continued the successful manufacture of refrigerating machinery.

About 1897, spurred on by the opportunities which presented themselves, and his early training in the erecting of beet sugar factories, the Fred. W. Wolf Company, under the direction of Mr. Wolf, installed a number of beet sugar factories in Michigan, which were very successful. On account, however, of the growth of the refrigerating industry, the manufacture of beet sugar machinery was abandoned about 1900, and since that time all the energies of the Fred. W. Wolf Company have been devoted to the building and installing of the Wolf-Linde refrigerating and ice-making systems. Over 1200 machines have been built and erected under the direct supervision of Mr. Wolf, and he, without doubt bears the distinction of being the father of the refrigerating industry in the United States.

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2 Titles of papers, where placed after the name of an author, and appearing in their exact form, are in italic. Papers are indexed not under their titles, but under their subject matter.

3 The Society is not responsible as a body for the statement of facts or opinions in the papers and discussions.

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